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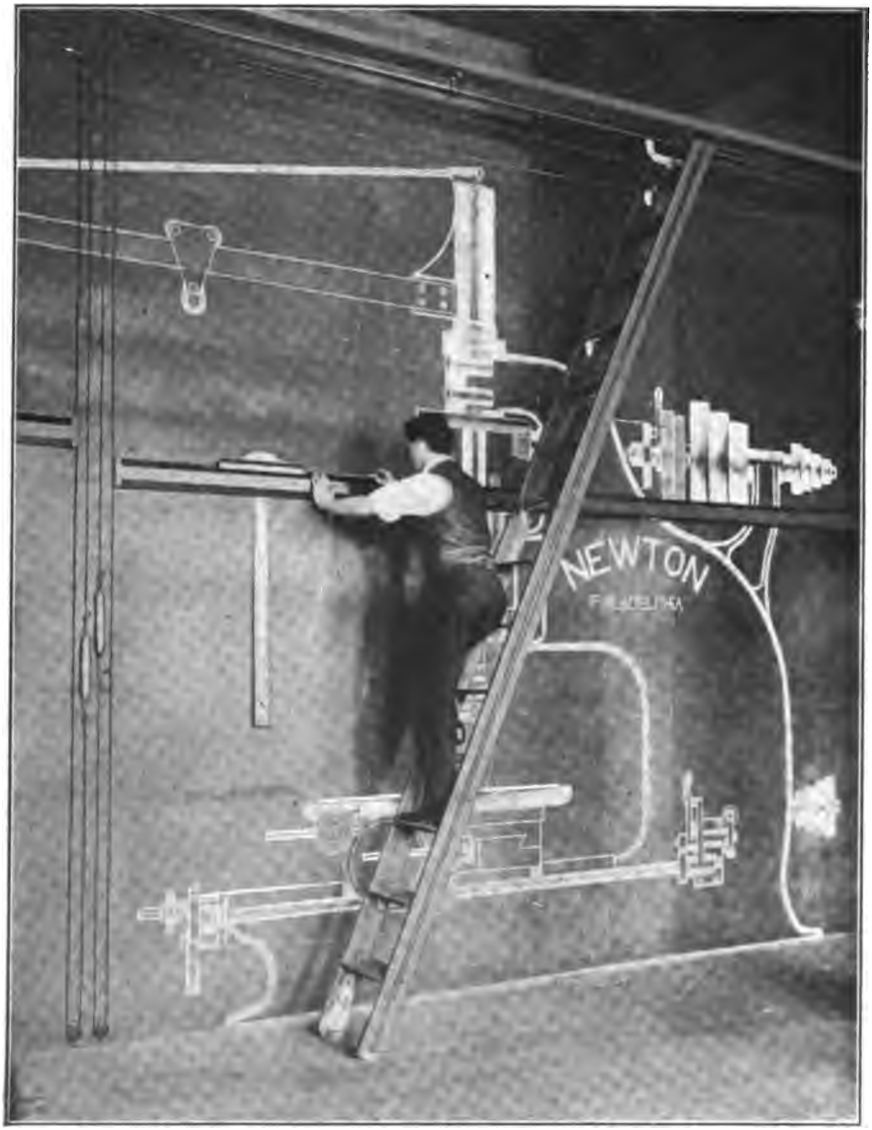

LARGE BLACKBOARD USED IN THE DRAWING ROOM OF THE NEWTON MACHINE TOOL WORKS.

Cyclopedia
of
Drawing

A General Reference Work

ON DRAWING AND ALLIED SUBJECTS FOR ARCHITECTS, MECHANICAL
ENGINEERS, DRAFTSMEN, CARPENTERS, STONE CUTTERS,
SHEET METAL WORKERS, TEACHERS, STUDENTS,
AND ALL OTHERS INTERESTED IN DRAWING.

Editor-in-Chief

ALFRED E. ZAPF, S. B.
SECRETARY, AMERICAN SCHOOL OF CORRESPONDENCE

Prepared especially for Practical Men by a Staff of

ARCHITECTS, DESIGNERS, AND EXPERTS OF ACKNOWLEDGED
PROFESSIONAL STANDING

FOUR VOLUMES

*One thousand seven hundred pages, two thousand illustrations,
tables and formulae*

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1907

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Foreword

THE Cyclopedia of Drawing has been prepared with the special object of giving the beginner and the self-taught practical man, a working knowledge of the principles which underlie all branches of drawing so he may know how to read and make drawings intelligently.

¶ The importance of drawing in the general plan of education is receiving wider and wider recognition. Few are the public schools which do not now teach it even in the primary departments. It is the universal language in which the peoples of all lands may communicate with each other,—and it is essentially the language of the architect and the engineer. Through it he communicates his noble designs, or his wonderful inventions, to his workmen for execution. As a cultural study drawing is of great value to every person who would lay any claim to a liberal education, teaching symmetry, beauty and exactness, and training the eye not only to see but to observe.

¶ A broad knowledge of drawing is therefore of vital importance to the draftsman, mechanic or young engineering student who is ambitious to advance in his chosen field. Unfortunately class room or even correspondence instruction is in many cases out of question, and thus many promising young men are left entirely without the means of gratifying their ambition.

¶ It is primarily for this class that the Cyclopedia of Drawing was published. It is based on the methods which the American School of Correspondence has developed and employed so successfully for many years in teaching drawing. It is compiled from the most valuable instruction papers of the School, selected, arranged and especially prepared for home study by a staff of experts and practical men, each an acknowledged authority in his specialty.

¶ The Cyclopedia has the unique advantage of having been revised and critically tested by actual use before being published, the individual instruction papers from which it is compiled having been studied and criticised by thousands of students in all parts of the world before being used in this work. In this way every obscure point, every small error so usual in technical books, has been discovered and corrected and many valuable practical suggestions incorporated.

¶ Parts I and II are devoted to the artistic or architectural side of drawing, Part III to drawing for engineers and mechanics, and Part IV, to drawing for sheet metal workers.

¶ The work as a whole is intended to form an authoritative, ready reference work on the broad general subject of drawing, and as such should prove a valuable acquisition to the general as well as the technical library.

¶ Each section has been prepared by an authority on that particular subject,—practicing architects, practical engineers, and teachers in the foremost engineering schools.

¶ Every section is profusely illustrated by drawings, sketches and valuable tables prepared especially for this work, and at the end of each section there is an examination to test the reader's knowledge, thus combining the advantages of a text book with a reference book.

¶ The instruction papers comprising the various sections are presented in exactly the same form as they are used in actual instruction as the purpose of the work is to bring to men who cannot take a correspondence course some of the benefits of the American School of Correspondence instruction, and through them to acquaint the public with the method and high standing of its instruction.

¶ In conclusion grateful acknowledgment is due to the staff of authors and collaborators. Without the hearty co-operation of these men of wide experience and acknowledged ability, this work would have been impossible.

THIS VOLUME CONSISTS OF FIVE OF THE FIFTY REGULAR INSTRUCTION PAPERS IN THE MECHANICAL ENGINEERING COURSE OF THE AMERICAN SCHOOL OF CORRESPONDENCE, ARRANGED IN CONVENIENT FORM FOR READY REFERENCE BUT NOT IN THE ORDER USUALLY STUDIED.

THESE INSTRUCTION PAPERS ARE PREPARED BY ACKNOWLEDGED AUTHORITIES AND REPRESENT YEARS OF PREPARATION TO ADAPT THEM TO HOME STUDY.

THE INSTRUCTION PAPERS OF THE AMERICAN SCHOOL OF CORRESPONDENCE ARE NOT FOR SALE TO THE PUBLIC. THE PRESENT EXCEPTION IS MADE FOR THE PURPOSE OF ACQUAINTING PEOPLE INTERESTED IN DRAWING WITH THE THOROUGHNESS OF THE INSTRUCTION OFFERED, IN THE HOPE THAT THE OPPORTUNITY FOR PERSONAL EXAMINATION THUS OFFERED WILL LEAD THE READER TO CONTINUE HIS STUDIES IN THE SCHOOL.

ALTHOUGH REPRESENTING ONLY A SMALL PORTION OF THE COMPLETE COURSE, IT IS CONFIDENTLY BELIEVED THAT THIS VOLUME HAS SUFFICIENT MERIT IN ITSELF TO MAKE IT OF IMMEDIATE VALUE TO EVERYONE INTERESTED IN ANY FORM OF DRAWING.

EXAMINATION QUESTIONS

FOLLOWING EACH SECTION ARE THE QUESTIONS OR PLATES WHICH CONSTITUTE THE REGULAR EXAMINATION OF THE AMERICAN SCHOOL OF CORRESPONDENCE. THEY OFFER THE READER A MEANS OF TESTING HIS KNOWLEDGE OF THE SUBJECTS TREATED.

INABILITY TO ANSWER THESE QUESTIONS, OR TO SOLVE THE PROBLEMS, WILL SERVE TO SHOW THE NECESSITY FOR FURTHER STUDY.

THE READER IS URGED TO SOLVE EVERY PROBLEM, CHECKING HIS RESULTS WHEREVER POSSIBLE WITH SIMILAR PROBLEMS IN THE PRECEDING PAGES. THIS WILL AFFORD AN EXCELLENT MEANS FOR FIXING THE MATTER IN HIS MIND.

STUDENTS PREPARING FOR COLLEGE OR CIVIL SERVICE EXAMINATIONS WILL FIND THESE QUESTIONS OF GREAT VALUE.

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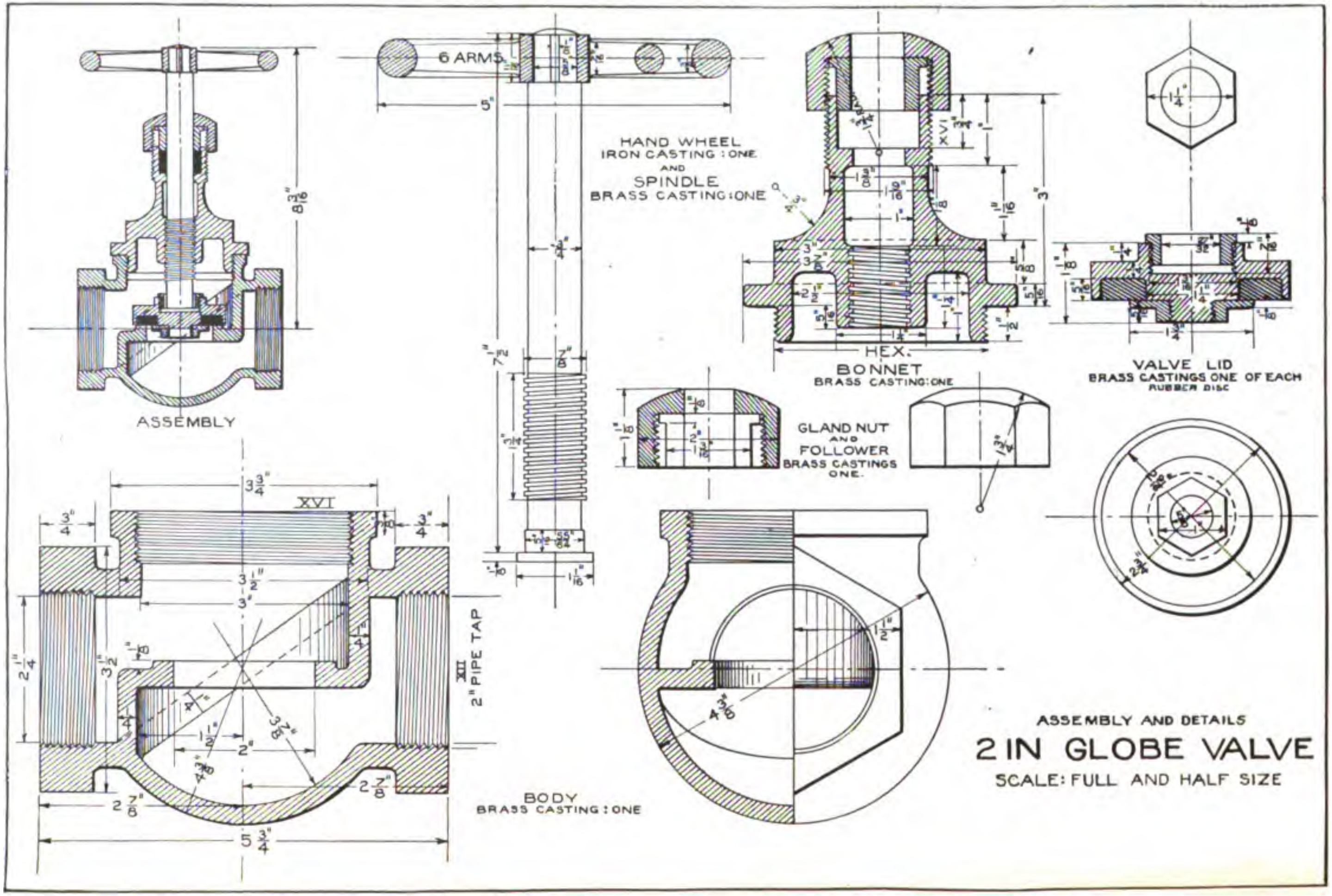


A TYPICAL WORKING DRAWING.

MECHANICAL DRAWING

PART IV

WORKING DRAWINGS

In Mechanical Drawing, Parts I, II, and III, the common drafting instruments and materials are described, and hints given regarding their use; the fundamental geometrical problems are solved; the principles of orthographic projection are stated, and their application to intersections and developments illustrated. A careful study of these parts, with the actual drawing work incident thereto, should have given the student considerable facility in producing good line work; he should now be able to draw neatly and accurately any simple piece which may be given him, correctly applying the principles as described.

In producing working drawings the principles already laid down are constantly used, and the more they are at the finger ends of the student the easier his work will become. The principles of projection must be thoroughly understood and fixed in the student's mind in order that he may devote himself with the greatest application to the actual detail of the drawing, and he must not be compelled at every step to turn back to find out how to make the simple projections.

Definition of Working Drawings. A working drawing is a drawing which completely instructs the workman, so that he is able to actually make in the shop the object which the drawing represents; in other words, a working drawing conveys to the mechanic all the information necessary to make the object. The student should constantly keep before him the idea that the workman must take a drawing, and, without any further instructions verbal or written, produce the object as the draftsman intended it to be made. The instruction supplied by the drawing should not only cover the form and size of the object, but also the kind of material of which it is to be made, the number of pieces desired, and the finish of its surfaces. A drawing, therefore, is a sort of abbreviated language, or shorthand method of conveying an amount of exact, detail information, which it would take many pages of manuscript to convey.

A second point to be noted in connection with a working drawing is that the workman has no time to puzzle over a mass of lines and figures more complicated than necessary. This means that special attention must be paid to making the drawing as simple as possible; all lines and figures which are unnecessary, beyond the point of conveying complete information, are hindrances rather than helps to the workman; moreover, it takes the draftsman's time to make these extra lines and figures, and thus the drawing becomes unnecessarily expensive. A good drawing, therefore, not only implies *accuracy and completeness* but also *simplicity and directness.*

Lines. The secret of a clear drawing, as far as the line work is concerned, lies not only in absolute uniformity in the making of the lines, but in choosing certain characteristic lines to convey different ideas. The most common kinds of lines used are described below, and the purposes of their use stated. They are also illustrated in Fig. 1.

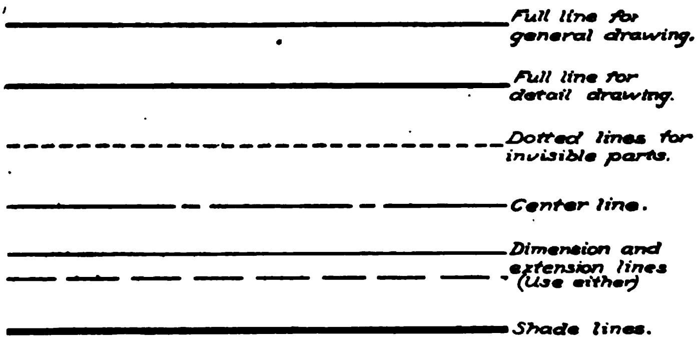

Full line for general drawing.
Full line for detail drawing.
Dotted lines for invisible parts.
Center line.
Dimension and extension lines (Use either)
Shade lines.

Fig. 1. Conventional Lines Used in Drawing.

Full lines represent the portions of the object which are visible; they should be bold and clear, heavy on detail drawings, say $\frac{1}{32}$ inch wide, and lighter on an assembled drawing.

Invisible lines represent the hidden parts of the object; they consist of short dashes regularly spaced, the spaces being about $\frac{1}{4}$ the length of the dash; the dashes should never have a greater width than that of the full line, and usually should be slightly less. A drawing is much easier to read if the full lines force themselves on the eye, while the dotted lines, by their lighter character, are left in the back ground.

Center lines or axis lines consist of alternate long and short dashes, finer than the main lines of the drawing. Some draftsmen prefer not to use "dash and dot" center lines, but make them continuous fine lines. Either style is good.

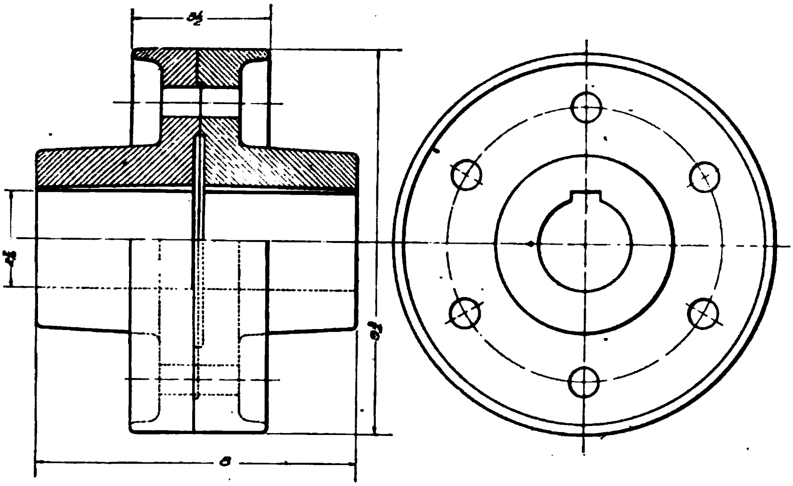


Fig. 2. Flanged Coupling Giving Practical Application of Fig. 1.

Dimension and extension lines are made fine, like center lines, and may be either full or dotted, according to the preference of the draftsman; the full line is preferable on account of its bolder character and the shorter time it takes to make it.

Extension lines start a short distance away from the edges of the object, so as to break up the continuity of the lines of the object and the extension line.

Dimension lines are run between the extension lines, terminating at the extension lines in arrows. The extension lines should always run a short distance beyond the point at which the dimension line touches them.

Shade lines are used for the purpose of more clearly bringing out to the eye the projecting edges of the object on the shadow side, and should be the heaviest lines on the drawing; the proper effect is secured if these lines are made nearly twice as heavy as the principal lines of the drawing.

Fig. 2 shows a flanged coupling in which the lines given in Fig. 1 are applied to an actual problem. In the lower half of the elevation observe how the invisible parts are shown by dotted lines.

Arrangement of Views. Imagine a rectangular block placed within a glass box, and the surfaces projected to the top, front and right-hand side, as in Fig. 3; now open the box in the manner indicated in Fig. 4 and we have three views of the object on a plane surface, *i.e.*, the drawing paper of the draftsman. These views are called top plan, front, and side elevations respectively, and are denoted in the figure by the letters T, F, and S. If more views are required, the arrangement is shown in Fig. 5. The bottom plan "B" is found below the front elevation, and the left side elevation S′ is found on the left of the front elevation, the same principles of projection being used as in the former case.

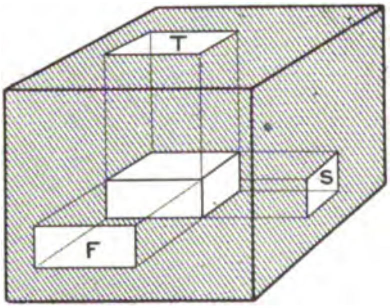


Fig. 3. Rectangular Block within a Glass Box, the Surfaces Projected to the Top, Front, and Right-hand Side.

The above procedure is equivalent to tracing on each side of the box the outline of the object as observed by the eye, when directly in front of each side of the object; after this is done the unfolding of the box results in the outlines shown in Fig. 4.

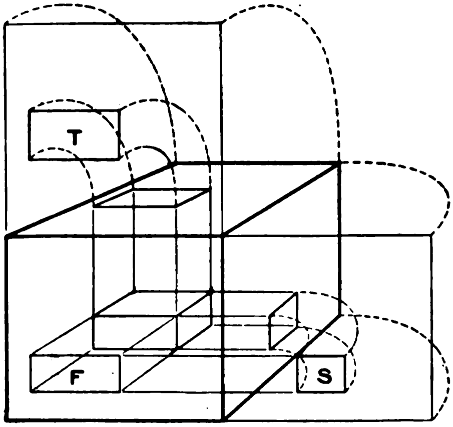


Fig. 4. Preferred Method of Showing Top, Front, and Right-hand Side Views of a Rectangular Block on a Plane Surface.

If we consider the front elevation of the object as our starting point, then the *top plan* is *above*, the *bottom plan below*, the view of the *right-hand side* is on the *right* of, and the view of the *left-hand side* is on the *left* of the front elevation. This arrangement of views is easily remembered and is very logical; it is the most common method of projection for mechanical drawing, and will be used throughout this book. For such a simple object as that considered above, two views only are necessary, a front elevation and top plan, but machine drawings frequently require three views, top, front, and side, and sometimes more.

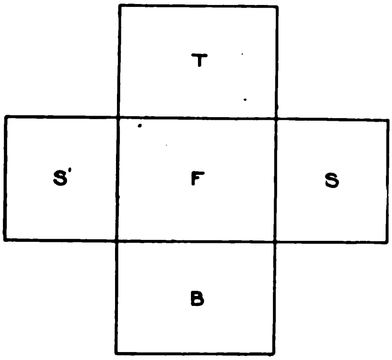


Fig. 5. Method of Showing Five Views of a Rectangular Block on a Plane Surface.

Some draftsmen prefer the method of projection shown in Fig. 6, by which the lines of the object, instead of being observed through an imaginary glass partition and traced thereon, are projected away from the eye upon surfaces *beyond* the object; the surfaces are then unfolded as before, with the result as shown in Fig. 7 that, the front elevation, being the starting point, the *top plan* is *below*, the *bottom plan* is *above*, the *left-hand* view is at the *right* of, and the *right-hand* view is at the *left* of the front elevation. This system of projection has few advantages for machine drawing, and has been largely superseded by the former method.

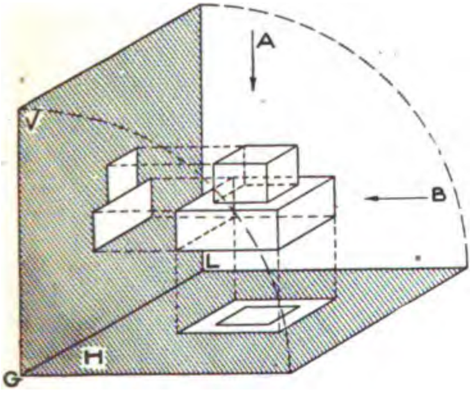


Fig. 6. Second Metohd of Showing an Object and its Projections in Vertical and Horizontal Planes. (Not Advised for Machine Drawing.)

Sectional Views. The interior construction of machine parts, especially if at all complicated, can seldom be clearly or completely shown by dotted lines. A large number of dotted lines on a drawing is very confusing, and in many cases renders the drawing useless. Sectional views are used to overcome this difficulty, and as an unlimited number of sections can be taken, it is always possible to make clear the interior construction of any piece, however complicated.

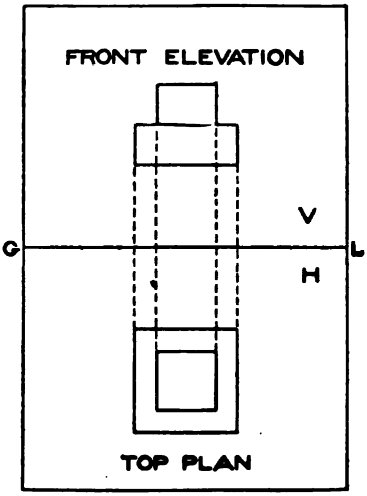


Fig. 7. Second Method of Projection, Planes Unfolding. (Not advised for Machine Drawing.)

To make a sectional view, the object is supposed to be cut open, and all the material removed between the cutting plane and the eye. This makes visible the hidden portion, and the drawing, therefore, consists of full lines made the same as any other, except that the material which was cut by the plane is "crosshatched." *Crosshatching* consists of drawing medium width lines, regularly spaced, across the cut surface, the lines usually being at an angle of 45° with the horizontal. In case of two adjoining surfaces being cut, the lines are sloped to the right and left respectively.

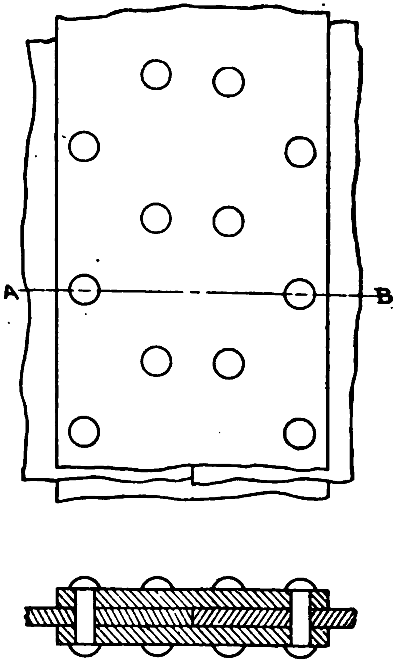


Fig. 8. Butt Joint Showing Use of Crosshatching when Section is Through Different Pieces of one Material.

The butt joint given in Fig. 8 shows the use of crosshatching when the section taken is through different pieces of the same material. Notice the different angles at which the section lines are drawn for each separate piece.

It is often convenient to show the kind of material of the object by the style of crosshatching. The conventional styles usually used are illustrated in Fig. 9. It is quite general, however, to use the plain form (as for cast iron), and call for the material by a specific note, thus leaving no possible doubt of the material required, and simplifying the labor of crosshatching, which is a tedious process at best. The distance between the lines should be as wide as possible, to save labor, and yet bring out the surface clearly. A good average spacing is about $\frac{3}{32}$ inch. Fig. 10 shows the end of a connecting rod. The section shows the different materials of which the object is made, wrought iron, brass, and babbitt.

Shade Lines. The theoretical principles for shade lines, already given in Mechanical Drawing, Part III, cannot be exactly applied to working machine drawings without involving an excessive amount of time and labor. The conventional rule, therefore, has been established that *shade lines may be used for all lower and right-hand projecting edges.* By "projecting edges" is meant edges of surfaces which are not flush with adjoining surfaces, but which project above them, or are in a plane nearer the eye. All views of an object are treated alike, the ray of light casting the shadow being supposed to come over the left shoulder of the draftsman as he faces the view on the paper. The contour lines of cylinders, cones, and other rounded surfaces, *if projecting,* are shaded the same as sharp edges.

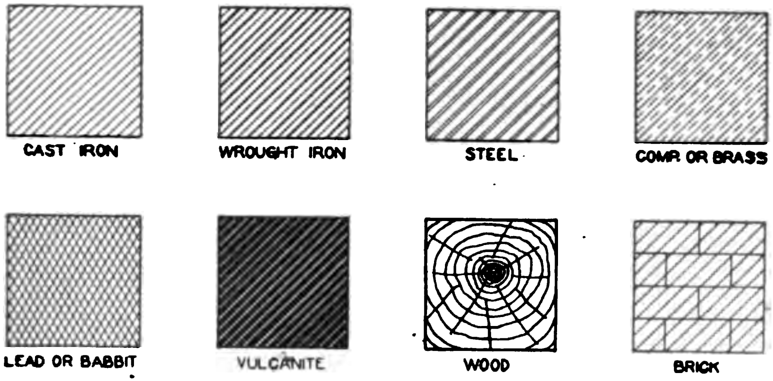


Fig. 9. Conventional Representation of Materials.

Shade lines, when used, are for the specific purpose of relieving the flatness of a drawing, and represent a purely conventional means of indicating to the eye projecting surfaces, *i.e.*, surfaces which are in different planes parallel to the eye. Whether the surfaces be curved or flat, as long as they are projecting, or in front of other surfaces, is of no moment, for the effect desired is the same for both, namely, the separation of the surfaces.

Few drawing offices allow shade lines to be used on regular detail machine work on account of the extra labor required and the loss of accuracy in the drawing by the use of a wide line. For general or "show" drawings, where the effect of separating the surfaces is desired, thus bringing out more clearly the relation of the parts, the use of shade lines is occasionally permitted. The draftsman should know how to apply them when required, and Figs. 11 to 16 illustrate their use.

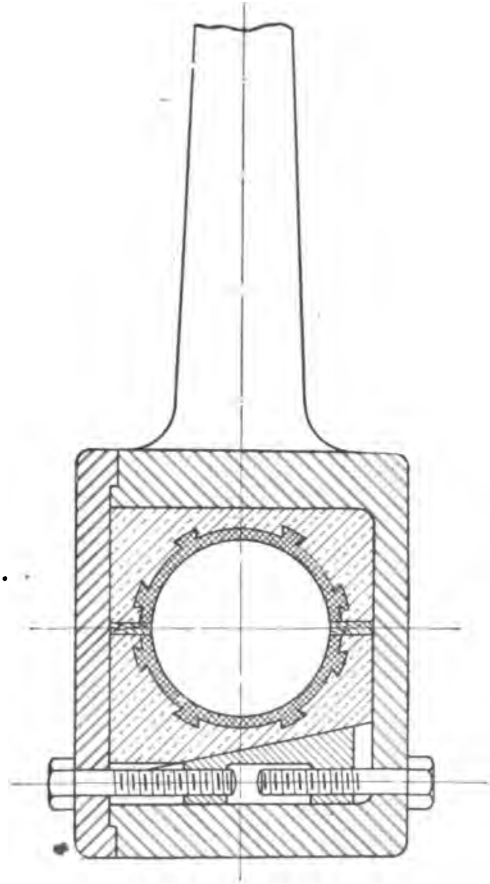


Fig. 10. End of Connecting Rod Showing Crosshatching when Section is Through Different Materials.

Fig. 11 shows the assumed direction of the ray of light in each of the three views. The piece should be inked in with the usual standard width of line, then gone over the second time, making the extra width for the shaded lines on the *inside* of the proper lines. This leaves the outside measurement of the piece unchanged for possible scaling. Some draftsmen claim that they can make the heavy shade lines as they go along, thus avoiding the second inking, but in the long run it will be found that time will be saved, more uniform lines and fewer blots made, if the process of shading be accomplished by a second inking.

Fig. 12 is similar to Fig. 11 but with a hole instead of a lug, and the difference in shade lines should be noted.

Fig. 13 is the same as Fig. 11 with a round boss and the lower right-hand corner rounded.

Fig. 14 is a plain washer, Fig. 15 a common hexagonal nut.

Fig. 16 is a washer or disc with a shaft in it. The right and left-hand views are shown to bring out the point that the shaft projecting on the right has its end shaded, while on the left, being flush with the face of the disc, it is not shaded.

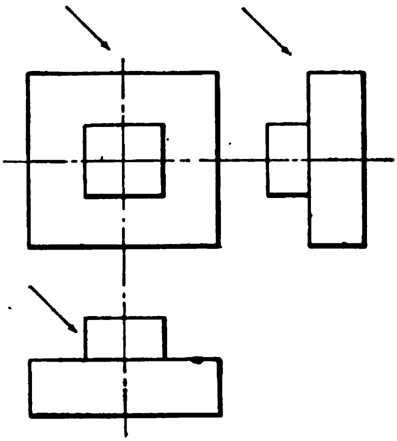


Fig. 11. Direction of Light Rays and Location of Shade Lines.

Dimensions. It is easy to spoil an otherwise good drawing by loose and careless methods of putting on dimensions. Systematic and careful effort must constantly be used to make every dimension upon a drawing absolutely clear. To put it still more strongly, it must be absolutely impossible for any dimension to raise doubt in the workman's mind as to its meaning. The draftsman has no justifiable excuse for mistakes in the shop due to poorly made dimension lines or small and blotted figures.

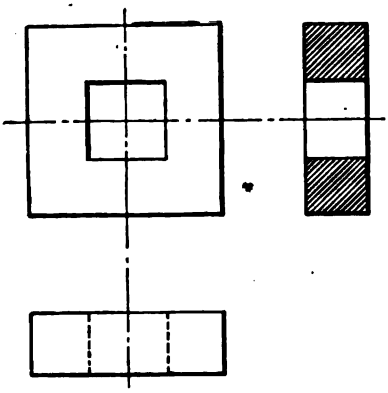


Fig. 12. Direction of Light Rays and Location of Shade Lines.

The arrows terminating the dimension lines should be pointed, bold, and regular, thus |←——3½——→|, not like this, |<——3½——:. The arrow points should exactly touch the extension lines, thus, |←—$5\frac{1}{4}$—→|, not like this, |←—$5\frac{1}{4}$—→|. The figures should be broad, bold, and clear, and of good size to be easily read. A gap may be left for the figure, thus, |←—$3\frac{5}{8}$—→|, or the line may run straight through, thus, |←—$3\frac{5}{8}$—→|, or the figures may be placed wholly above the line thus, |←—$3\frac{5}{8}$—→|.

It should be noted that by making figures broad, they will appear bold and clear, even when they are limited to small height. The common error of making them narrow destroys their bold character, and renders them difficult to read. Note the difference between the following examples, both sets of figures being exactly the same height, but one broad and the other narrow, FIG.25 FIG.25.

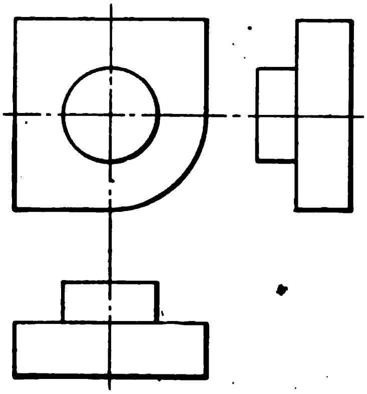


Fig. 13. Direction of Light Rays and Location of Shade Lines.

Fractions should always have the dividing line horizontal, thus, |←—$\frac{11}{16}$—→| not like this, |←—11/16—→|.

Small dimensions, or dimensions in cramped places should be made thus, →| |←$\frac{3}{16}$ or →| |←$\frac{1}{8}$ or →|$\frac{1}{8}$|←.

Dimensions in feet and inches are usually given for distances above 36 inches; but often for distances above 12 inches, and should be indicated as follows, the dash being made bold and conspicuous, |←—4'-6"—→|.

MEANING OF COMPLETE INSTRUCTIONS

It is naturally difficult for the student to determine what constitutes "complete instructions to the workman," and this knowledge can only be *fully* acquired by experience, both in drawing room and shop. Association, however, with the shop men who use drawings, a careful observation of their operation of tools, and a general familiarity with handling of material in a shop help wonderfully in getting the right point of view and proper spirit for making a good drawing. When one stops to think about it, to give instructions without having the least idea of how the workman will go about it to follow them seems the height of foolishness, yet that is what the student who tries to make working drawings wholly from book rules is doing. He should use his book knowledge as a guide and constant help, but he should be a "shop man" *first, last, and all the time*. When he has acquired the habit of constantly putting himself in the workman's place his drawings will be all right, they will convey "complete instructions to the workman."

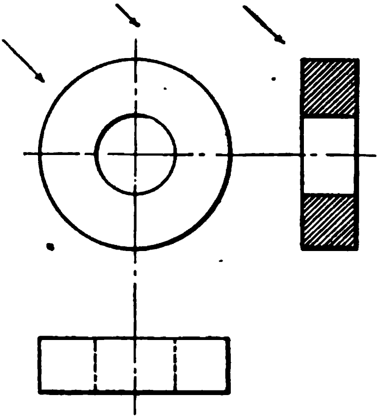


Fig. 14. Direction of Light Rays and Location of Shade Lines.

In the ordinary run of shop work there are several classes of workmen who have to use drawings. Broadly classed, they are as follows:

Pattern-makers,
Blacksmiths,
Machinists (including Toolmakers, Special Machine men, and Erectors).
Order and Receiving Clerks.

These several workmen will use the same drawing, and the instruction which it conveys must be so arranged that each can readily pick out the portion which he needs to enable his work to be properly done. The general requirements of each are discussed below, and form the basis for the style and methods of dimension-

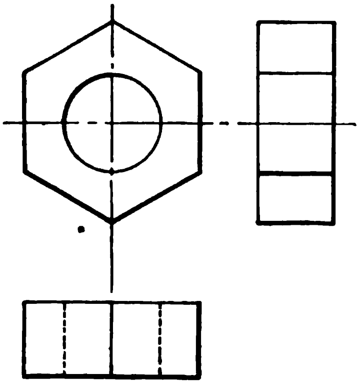


Fig. 15. Direction of Light Rays and Location of Shade Lines.

ing drawings used in common practice, and illustrated in this book.

The *pattern maker*, on receiving a detail drawing of a piece, first proceeds to copy it full size, divided up into such sections as are convenient, upon his work board. This board is merely a large smooth table top, set up on a couple of horses. Sometimes brown paper is tacked on this board and the pattern drawing made on it, but more often the pencil lines are made directly on the surface of the board and the board resurfaced for future work. He does not make a finished drawing, but with his straight-edge, large dividers, and compasses he lays out enough to enable him to see and measure the detail at all points.

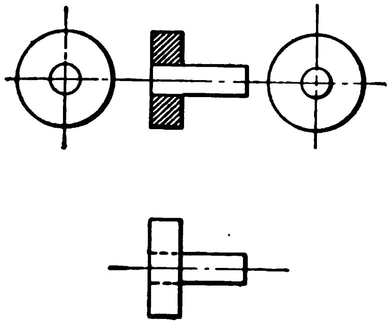


Fig. 16. Direction of Light Rays and Location of Shade Lines.

This pattern layout is made for a number of reasons. Molten cast iron, when it cools, shrinks about $\frac{1}{8}$ inch per foot, so the pattern has to be made larger than the figures on the drawing call for. In order to save calculation for each dimension, a "shrink rule" is used, each foot of which is made $12\frac{1}{8}$ inches long. The pattern maker uses this rule in all his work, and thus makes his layout on a "shrink" basis.

Wherever the drawing calls for finished surfaces, the stock of the pattern has to be increased by $\frac{1}{8}$ inch or more, and this addition has to show on the pattern maker's drawing. In order to get the casting out of the sand of the mold, "draft," or taper on the pattern, has to be allowed. As the draftsman cannot always predict just how the piece will be molded, the "draft" is not shown on the office drawing, and the pattern maker, therefore, has to make the allowance and show it on his drawing. All fillets, sizes of cores and core prints, details of core boxes and loose pieces of the pattern have to be carefully worked up on the pattern drawing. The result of this special pattern layout often is that certain minor changes have to be made in the shape and size of the piece to permit the pattern to be properly built and readily molded in the foundry, for a good pattern maker has also to be a good foundryman. We thus see that, as far as the pattern maker is concerned, the drawing must be very complete as to detail, both inside and out, and carry dimensions for all surfaces, cores, fillets, corners, etc.

These dimensions must be in even figures as far as possible, as a pattern maker's rule seldom reads finer than $\frac{1}{16}$ inch. Gear work is a specialty, and decimals are allowable, and there are certain other cases where odd dimensions cannot be avoided. In arranging the dimensions on the drawing, the more knowledge the draftsman has of pattern making, the more conveniently will he figure the drawing for the pattern maker. He will, in figuring the interior of a casting, think of the core box which will be made, and will be sure that he gives the length, breadth, and depth of the cavity, and all corners, bosses, and lugs projecting into it, with simplicity and clearness. He will give dimensions for all sloping lines and odd shaped curves definitely and carefully, thinking all the while of the pattern maker and his tools, the square, straight-edge, dividers, and compasses. He will avoid thin edges, and frail projections, and awkward intersections. The consideration of such points as these is what makes a good drawing for the pattern maker to use, and much reduces cost of the pattern, for pattern-making is relatively high priced labor. Some of these points are really points of machine design, but it is not possible for a good detail drawing to be made without using to a small extent, at least, the elementary principles of design. These the student can unconsciously acquire by familiarizing himself with actual shop work.

The *blacksmith* sometimes uses a pattern for a forging. This is to enable him to lay aside the drawing for pieces which are to be made in large numbers, and set his calipers quickly and accurately from a pattern. Simplicity of shape is of even more importance to the blacksmith than to the pattern maker. The stock material of the blacksmith consists of straight bars of iron and steel of round, square, and oblong cross-section. All bosses, lugs, hubs, or sudden variations of shape have to be produced by "upsetting" or welding, either of which is a process involving time, care, and expense. Forgings, therefore, should be, as far as possible, simple, straight pieces, with few bosses or lugs, and when bends are necessary they should be of the simplest nature.

In forging a piece the blacksmith has to work quickly, and has no time to read or measure fine dimensions, it is therefore useless to expect him to read any finer dimensions than $\frac{1}{16}$ inch; special attention should be paid to giving him *overall* dimensions, not only for cutting off the stock, but for enabling him to quickly make his measurements while the piece is hot and gripped by his tongs on the anvil. The blacksmith has to make about the same allowance of extra stock for finished surfaces as the pattern maker.

The *machinist* uses only a few of the figures on the average drawing, while the pattern maker and blacksmith use practically all of them. The machinist is concerned only with finishing the piece, and views the drawing with regard to the machine work upon it. In order to finish the surfaces accurately in proper relation to one another it is necessary to choose some fundamental face of the piece, first finish that, and then use it as a gauging surface to work the others from. The draftsman, if he is reasonably familiar with shopwork, can usually foresee what this gauging surface will be. This has an important influence on his dimensions, for he should so give the dimensions that the machinist will find them convenient and consistent with all his operations on the piece.

When special tools, jigs, fixtures, and automatic devices are applied for the finishing of pieces in large numbers, the method of dimensioning is apt to be somewhat different from that on the general run of machine work. A free use of notes on the drawing, specifying the nature of finished surface desired, or the kind of fit, or any special points in connection with the machining of the piece, is valuable to the machinist. It is not good economy to spend any more labor on securing a finished surface than the purpose for which it is made requires. For example, in turning up a shaft with a number of bearings along it, most of its surface being free, care should be taken to finish the parts for the *bearings* to an *exact* diameter, but for the balance of the length a smooth surface with the diameter reasonably accurate is all that is necessary. The drawing should specify this difference of finish so that the machinist will not waste time on the piece.

The special operations on a piece, such as cutting of gear teeth, grinding and "lapping" of shafts, cutting of threads, etc., are all done subsequent to the main finishing of the piece. For example, the casting for a cut gear is first bored, the hub faced, and the solid rim turned and faced to the dimensions on the drawing. This produces the "gear blank." The subsequent operations of cutting the teeth on an automatic gear cutting machine, and keyseating the hub on a keyseater, require additional instructions as to the style of cutter, number of teeth, dimensions and style of keyway, etc.

Machine shops are differently equipped for doing the same kinds of work, and this has an important influence on the manner of placing the finishing dimensions on drawings. Thus, some shops have rotary planers instead of the regular reciprocating platen type. Some have turret lathes, screw machines, and horizontal boring mills, while others have only lathes. Some have grinding machines, both for flat surface and cylindrical work, the final finishing cuts being taken on these machines, after the pieces have been roughed out on the lathe and planer. Grinding machines are now regularly built to take a heavy cut and coarse feed for roughing out the work, thus often dispensing entirely with the lathe. Milling machines, though found in all shops, are used in widely different scopes. Some use the milling machine almost entirely, to the exclusion of the planer, specially heavy machines being adapted for this purpose. Locomotive shops differ in their equipment and practice from stationary engine shops; machine tool and automatic machinery builders have little in common with the equipment of shops for manufacturing heavy power transmission machinery; steam pump shops are wholly different in their equipment from that of an establishment building electrical machinery.

All these differences have an important bearing on the style of drawings needed, and especially so on the methods of giving the dimensions for the use of the machinist. Without attempting to give an idea of how to control each case, which would be well nigh impossible, suffice it to say that the student should become impressed with the fact that he must *study the workman constantly* in order to be able to give him upon the drawing the necessary "complete instructions."

The *order and receiving clerks* are very easily satisfied as far as their parts in the use of a working drawing are concerned. They simply need a designating mark, a pattern number for a casting, and a piece number for all other parts, together with the material and number wanted of each piece, in order that the proper orders may be written, and the material identified for recording its receipt in the shop or field. The number wanted is often given only on a separate "bill of material" which accompanies the drawing, but it is also quite general to note on the drawing in the title of each piece, the number wanted for one complete machine, whether billed elsewhere or not.

Screw Threads. Exact drawings for the helix forming the thread of a screw are shown in Mechanical Drawing, Part V. These not only are difficult to draw, but they consume considerable time to produce accurately, therefore draftsmen have adopted certain conventions to represent the thread on working drawings. Some of these conventions are shown in Fig. 17. Here "a" represents a single, right-hand, square thread; "b", a single, right-hand, sharp V thread, and its modifications, the U.S. Standard or Seller's thread, and the Whitworth thread; "c" represents a left-hand, sharp thread; "d" is the most common convention for any thread of a V shaped cross section; "e" for any thread on a very small bolt or set screw; "f" is a modification of "d", there being no slope to the thread, which convention is preferred by some draftsmen; "g" represents a standard pipe thread, the taper on the sides of the pipe being neglected.

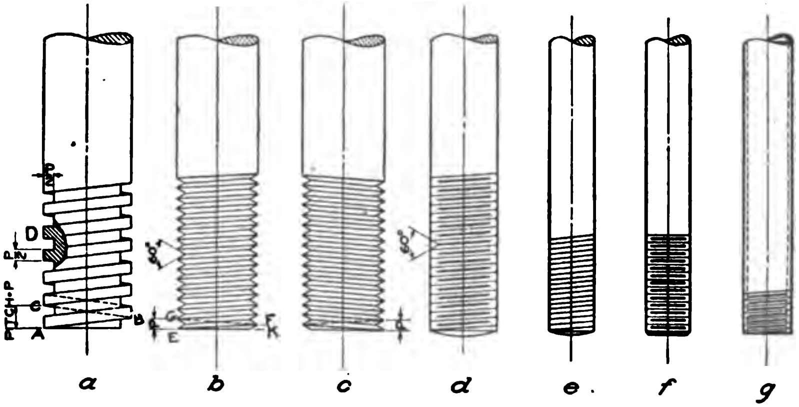


Fig. 17. Conventional Representations of Screw Threads.

There are other conventions for threads in use, but the above are the most important ones. These certainly can not be mistaken for anything else, which is the real test for any conventional representation of an object.

The pitch of a screw thread is the distance between corresponding points on two successive threads measured parallel to the axis. A small axial section is shown at "D" on the thread "a." The square groove, which gives the thread its name, has a depth equal to about ½ the pitch. Starting at the bottom, and following the edge of a thread in making *one turn* around the bolt, or from A to B on the front and B to C on the rear, we find that the thread advances parallel to the axis a distance AC, or the pitch. As we can see but one half of a turn it will be noted that a single right-hand thread advances a distance equal to ½ the pitch along the right-hand side, and similarly for a single left-hand thread, the distance advanced would be ½ the pitch on the left-hand side. The slope is, therefore, upward and to the right in the first case, and upward and to the left in the second case.

To draw the thread, space off the sides of the bolt with the dividers set to ½ the pitch, determine the slope, whether for single, double, or triple thread, and fill in the tops of the threads. The depth of the thread, say ½ the pitch, should then be laid off on each side and the lines drawn which show the visible portion of the bottom of the thread, thus completing the view for the ordinary convention. When the slope is considerable, as in this case, a small portion of the rear thread becomes visible, and may be shown.

For the thread at "b" it will be readily seen that an axial section would give V grooves, and as the standard angle for the grooves of the thread in this country is 60°, the projections are equilateral triangles. As before, if we follow a thread around the bolt, or from E to F on the front, and F to G on the back, we find for the single thread screw that the visible portion of the thread EF advances along the right-hand side a distance equal to ½ the pitch. Hence starting at the bottom, laying off the distance KF equal to ½ the pitch on the right-hand side, and connecting E to F we have the slope of the thread determined. Spacing the pitch on the left-hand side for such distance as is required, we may then draw through these points lines parallel to EF, or the top lines of the thread. The Vs may then be drawn with the aid of a T square and 30° triangle, after which the bottom of the threads may be connected. It will be noted that the top of the thread on one side is directly opposite the bottom of the thread on the other side. The left-hand V thread at "c" is drawn in a similar way to "b," the thread advancing on the left-hand side instead of on the right.

Considerable care is required to get the Vs uniform, and the more practical, usual, and in nearly all cases satisfactory method of showing the thread is given in the remaining figures.

The spacing for the conventions "d," "e," and "f" approximates the pitch of the thread, and time can be saved by the draftsman learning to space with the eye rather than with the dividers. The light lines representing the top of the thread should be drawn first, the heavier lines for the bottom of the thread are then drawn midway between the light lines, stopping a short distance from the edges of the bolt. Often no difference in width is made between the lines representing the top and bottom of the threads, thus still further simplifying the conventional representation. It is well for the beginner to draw pencil lines limiting the bottom of the thread, so that the ends of the heavy lines will not be ragged or irregular. It rather improves the appearance of the thread to have the slant exaggerated in "d," "e," and "g."

Threads in Sectional Pieces. Figs. 18, 19, and 20 illustrate the common method of representing threads when they occur on pieces which are drawn in cross-section. The front half of the piece is supposed to be removed and we are looking at the back half. Now the thread on the back side of a screw slants the opposite way from what it does on the front side, and of course the same is true of the thread in a tapped hole. Consequently, since it is the back side of the hole which is seen, the slant of the lines which represent the thread is opposite to the direction they would have were we looking at the front side of the screw which goes into the hole. We have just learned that for a right-hand thread on a screw the lines slant upward from left to right, and therefore for a right-hand thread seen on the back side of a tapped hole, the lines will slant upward from right to left. In other words, for a right-hand thread in a hole which comes in a cross-section, the lines slant the same as they would on the front of a left-hand thread on a bolt; and for a left-hand thread in a sectioned hole, the slant is the same as for a right-hand thread on a bolt.

Fig. 19 is a piece which has a smooth hole through it and a thread on the outside. Here the entire thread is invisible, except at the contour of the cylinder, and must be indicated by the notches. These are drawn by spacing off the distance which is used for the pitch and from the points thus found drawing lines with the triangle which make an angle of 60 degrees with the axis of the cylinder. For a single thread the notches on one side have their outer points opposite the inner points of the notches on the other side. For a double thread the notches are directly opposite each other.

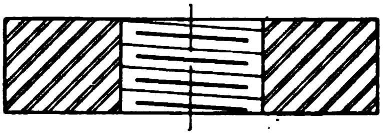


Fig. 18. Conventional Drawing for Threads in Sectional Pieces.

Fig. 20 shows two ways of quarter-sectioning a threaded piece, the only difference being that on one the contour of the sectional part is drawn a straight line, while on the other the contour is notched. Either one may be used. The straight contour can, of course, be drawn much more quickly and in places where there is no danger of sacrificing clearness it should be used for that reason. If the drawing is somewhat complicated, so that without the notches it might not be quite clear that the piece was threaded, the notches should be used.

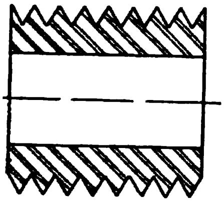


Fig. 19. Conventional Drawing for Threads in Sectional Pieces.

As has already been suggested, the student will doubtless find many other customs in the matter of drawing threads which are quite as good as the above. These have been given as ones which are common, and easily drawn. As a matter of convenience the following tables are given, which show the number of threads per inch on some of the most common sizes of bolts, according to the standard adopted by the United States Government, and the Whitworth or English standard.

UNITED STATES STANDARD SCREW THREADS

Diameter of Bolt.	Threads per Inch.	Diameter of Bolt.	Threads per Inch.	Diameter of Bolt.	Threads per Inch.
$\frac{1}{4}$	20	$\frac{5}{8}$	11	$1\frac{3}{8}$	6
$\frac{5}{16}$	18	$\frac{3}{4}$	10	$1\frac{1}{2}$	6
$\frac{3}{8}$	16	$\frac{7}{8}$	9	$1\frac{5}{8}$	$5\frac{1}{2}$
$\frac{7}{16}$	14	1	8	$1\frac{3}{4}$	5
$\frac{1}{2}$	13	$1\frac{1}{8}$	7	$1\frac{7}{8}$	5
$\frac{9}{16}$	12	$1\frac{1}{4}$	7	2	$4\frac{1}{2}$

WHITWORTH STANDARD SCREW THREADS

Diameter of Bolt.	Threads per Inch.	Diameter of Bolt.	Threads per Inch.	Diameter of Bolt.	Threads per Inch.
$\frac{1}{4}$	20	$\frac{5}{8}$	11	$1\frac{3}{8}$	6
$\frac{5}{16}$	18	$\frac{3}{4}$	10	$1\frac{1}{2}$	6
$\frac{3}{8}$	16	$\frac{7}{8}$	9	$1\frac{5}{8}$	5
$\frac{7}{16}$	14	1	8	$1\frac{3}{4}$	5
$\frac{1}{2}$	12	$1\frac{1}{8}$	7	$1\frac{7}{8}$	$4\frac{1}{2}$
$\frac{9}{16}$	12	$1\frac{1}{4}$	7	2	$4\frac{1}{2}$

Bolts and Nuts. A *bolt* is a cylindrical bar upset at one end to form a head and having a screw thread cut at the other end.

A *nut* is a hollow piece of metal in which a screw thread has been cut.

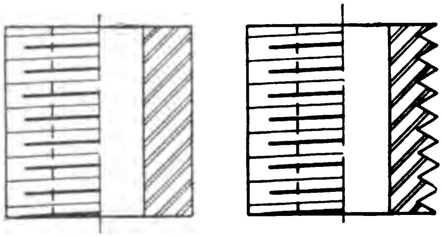


Fig. 20. Conventional Drawing for Threads in Sectional Pieces.

A right-hand bolt has its thread so cut that its nut goes on, or advances along the axis of the bolt, when turned in the same direction as the hands of a watch. A left-hand bolt has its thread so cut that its nut must be turned opposite to the hands of a watch in order to have it go on.

Figs. 21 and 22 show conventional drawings of a $\frac{3}{4}$-inch machine bolt having a hexagonal head and nut. The head is simply a hexagonal prism which has been chamfered, *i.e.*, the corners rounded off so that the top view shows a circle inscribed in a hexagon, see Fig. 23. This top view has been omitted in Figs. 21 and 22, the conventional drawing being considered sufficient to show that the head is hexagonal, yet some draftsmen prefer to specify whether

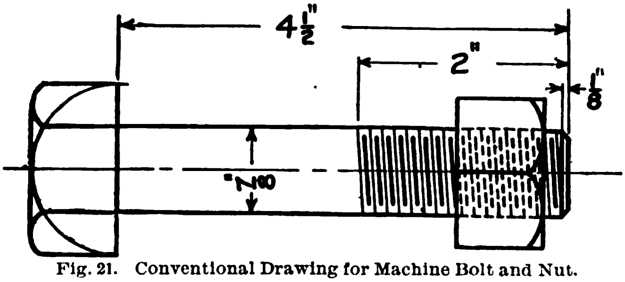


Fig. 21. Conventional Drawing for Machine Bolt and Nut.

the head and nut are hexagonal or square, thus, "SQ. HD.," "HEX. N.," as the case may be. The view across the *long diameter* is given in preference to the view across *the flats*, so that in close quarters the clearance of the corners may be readily seen. The shank of the bolt is represented as explained above for conventional threads. The point is chamfered a little in the figure so that it appears as the frustum of a cone.

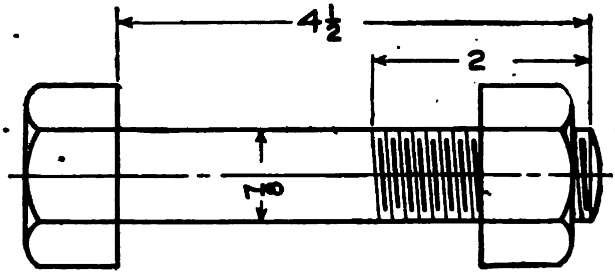


Fig. 22. Conventional Drawing for Machine Bolt and Nut.

Another style of point is shown in Fig. 24, where the end is rounded off with a radius equal to about 1½ times the diameter.

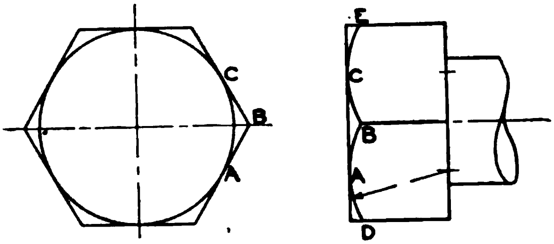


Fig. 23. Conventional Drawing for Machine Bolt Head.

The lines which represent the thread should not cross the line drawn square across the bolt where the chamfer or rounding of the point begins. Note that in Fig. 21 the threads have been shown dotted through the nut, while in Fig. 22 the simpler and more common method is followed of omitting the dotted threads and showing the long diameter of the nut.

The dimensions given are all that are necessary for the workman to make a *standard* bolt and nut. Some draftsmen prefer to show the length of bolt and thread from the base of the frustum or spherical end, as in Fig. 22, but this does not at once give the *total* length under the head, which is usually the important figure.

In case dimensions for the head are needed, the thickness and the distance across flats should be given, as obtained from the table of proportions on page 26. If standard threads are not used, then the number of threads per inch must be given.

A method for making a conventional drawing of a hexagonal bolt head or nut is shown in Fig. 24. From A as a center describe an arc with a radius equal to the diameter of the bolt, making it intersect the perpendicular to the center line through A at points B and C. Continue the sides of the bolt until they intersect the arc at E and F, and draw lines through B and C parallel to sides of bolt. Draw a tangent to the arc parallel to BC for the top of the head. Find by trial the radius G and draw the arcs for the sides of the head. It will be noted that the long diameter of the hexagon by this method is twice the diameter of bolt, which is practically true to the standard table for bolts under I inch diameter, and sufficiently exact for the larger sizes in common use.

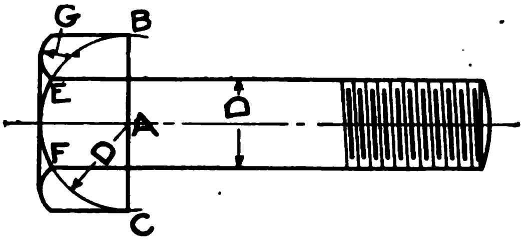


Fig. 24. Conventional Drawing for Machine Bolt.

Fig. 25 is a full size drawing of a 2½ inch rough bolt, having a hexagonal head and U.S. standard thread. Dimensions for the height and width of the head have been taken from table of bolt heads on page 26. The width of the head, 3⅞ inch, is the diameter of the chamfer circle and is the first portion of the plan view to be drawn. Then the hexagon is circumscribed about the chamfer circle. Project the width of the faces and B D for the flat portion of the top. Assuming the chamfer to be conical and at 45° with the axis, draw lines B C and D E. The curves of intersection are approximated by arcs of circles springing from E and C and drawn tangent to B D.

The enlarged thread section below shows that the thread is flattened at top and bottom by cutting off ⅛ of the depth F of the V thread. Note that the pitch of the thread is laid off on a line located outside of the true diameter of the bolt by an amount equal to the portion cut off the ordinary sharp V thread. The end is rounded with a radius equal to the diameter, or preferably 1½ times the diameter.

Two views, plan and elevation, of two faces of a hexagonal bolt head are given in Fig. 23. The chamfer circle is tangent to the sides of the hexagon, which means that the tool, in making the chamfer, cuts off the corners of the top as at A B C. The true curves D A B and B C E are lines of intersection of a cone or sphere with a hexagonal prism and may be easily obtained by the principles of projection. A simpler and much more convenient method is to approximate these curves with arcs of circles, using the height of the head as a radius, as shown.

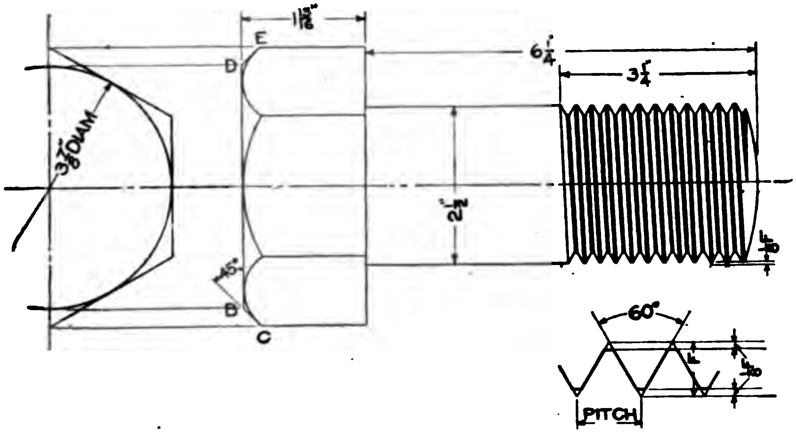


Fig. 25. Machine Bolt with U. S. Standard Thread.

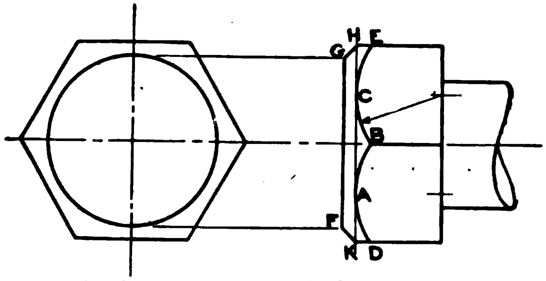


Fig. 26. Hexagonal Bolt Head with Chamfer.

In case it is desired to show more chamfer as in Fig. 26, the top of the head may be cut off at F K and G H at an angle of 30° or 45° and the diameter of the chamfer circle is projected to the plan view as shown. The width of the hexagon is the same as before and is readily projected from the plan view. The curves of intersection D A B and B C E are drawn as arcs of circles, but instead of being tangent to the top they are tangent to the line K H, which indicates the place where the chamfer cuts the flat sides of the hexagon.

Fig. 27 shows three views of a square head or nut with chamfer corresponding to that on the hexagonal head in Fig. 23: and Fig 28 shows the square head or nut chamfered to correspond to Fig. 26. Referring first to Fig. 27 the arc on the side view which shows the short diameter of the nut is drawn with a radius A, equal to two and one-quarter times the diameter of the bolt on which the head or nut belongs. The arcs on the other side view are drawn with a radius B, equal to one-half of A. The lines EF are drawn from points E tangent to the arcs, and it will be found that the points of tangency will come almost at the points where the arcs cut the lines CD. Points E are found by projecting from the plan view as indicated.

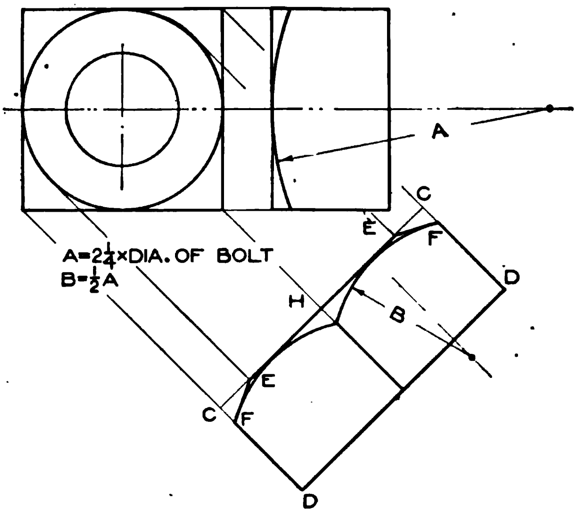


Fig. 27. Three Views of Square Head or Nut with Chamfer.

In Fig. 28, the construction is similar. The points N are first found by projecting from the top and bottom of the circle in the plan view; then the lines NL are drawn making angles of 30 degrees with line NN. (The proportions for the radii which are given, hold good only when the angle of 30 degrees is used.) Next draw the construction line LL and draw the arc tangent to it with a radius A equal to two and one-quarter times the diameter of the bolt, the same as in Fig. 27. To draw the chamfer in the other side view, draw the construction line parallel to and at a distance from CC equal to the distance LL from NN and draw the arcs tangent to this line with radius B equal to one-half of A. The lines EF are then drawn as explained for Fig. 27.

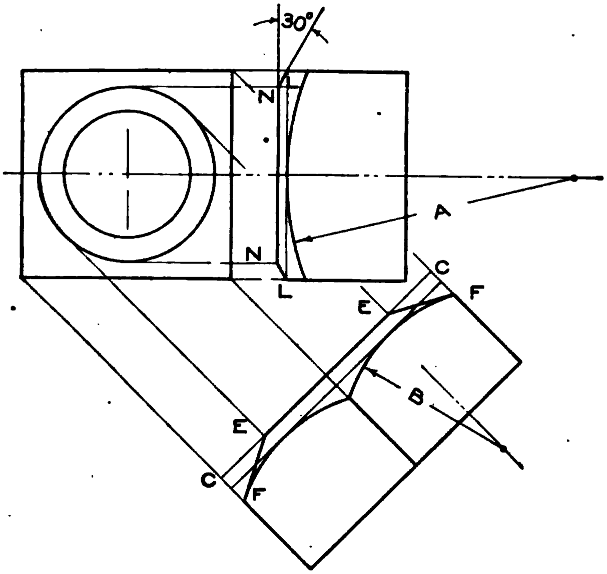


Fig. 28. Three Views of Square Head or Nut with Chamfer.

Most of the bolts in common use are made standard sizes, that is, for a certain diameter of bolt there is a corresponding standard diameter and thickness for the head and the nut, and a standard number of threads per inch, so that if the bolt which the draftsman wishes to use has these standard dimensions they may be omitted from the drawing and a note made that the bolt is standard. Then the only dimensions necessary to be given are the diameter, the length under the head, and the length of the threaded part.

The following tables give the United States standard sizes of square and hexagonal heads and nuts for bolts. The columns headed "Width of Nut" and "Width of Head" give the shortest dimension of the square or hexagon, that is, the diameter of the inscribed circle, or the distance across flats. The standard number of threads per inch can be found from the table already given.

ROUGH SQUARE AND HEXAGON BOLT HEADS.
U. S. STANDARD
(Franklin Institute)

Diameter of Bolt.	Width of Head.	Thickness of Head.	Diameter of Bolt.	Width of Head.	Thickness of Head.
1/4	1/2	1/4	1	1 5/8	13/16
5/16	19/32	19/64	1 1/8	1 13/16	29/32
3/8	11/16	11/32	1 1/4	2	1
7/16	25/32	25/64	1 3/8	2 3/16	1 3/32
1/2	7/8	7/16	1 1/2	2 3/8	1 3/16
9/16	31/32	31/64	1 5/8	2 9/16	1 9/32
5/8	1 1/16	17/32	1 3/4	2 3/4	1 3/8
3/4	1 1/4	5/8	1 7/8	2 15/16	1 15/32
7/8	1 7/16	23/32	2	3 1/8	1 9/16
			2 1/4	3 1/2	1 3/4
			2 1/2	3 7/8	1 15/16

ROUGH SQUARE AND HEXAGON NUTS. U. S. STANDARD
(Franklin Institute)

Diameter of Bolt.	Width of Nut.	Thickness of Nut.	Diameter of Bolt.	Width of Nut.	Thickness of Nut.
1/4	1/2	1/4	1	1 5/8	1
5/16	19/32	5/16	1 1/8	1 13/16	1 1/8
3/8	11/16	3/8	1 1/4	2	1 1/4
7/16	25/32	7/16	1 3/8	2 3/16	1 3/8
1/2	7/8	1/2	1 1/2	2 3/8	1 1/2
9/16	31/32	9/16	1 5/8	2 9/16	1 5/8
5/8	1 1/16	5/8	1 3/4	2 3/4	1 3/4
3/4	1 1/4	3/4	1 7/8	2 15/16	1 7/8
7/8	1 7/16	7/8	2	3 1/8	2
			2 1/4	3 1/2	2 1/4
			2 1/2	3 7/8	2 1/2

Pipes and Pipe Threads. The various kinds of pipe in common use are made to standard sizes, and as the draftsman very often comes in contact with piping we will consider it briefly. The kinds commonly used are wrought-iron or steel pipe, brass pipe made to the size of wrought-iron pipe, and cast-iron pipe. The cast-iron pipe is made of different weights and form, according to the purpose for which it is to be used. Standard wrought iron pipe is rated by its nominal inside diameter, although the actual diameter does not in most cases quite agree with the nominal diameter. For example, a ¼-inch pipe is a pipe, the hole in which is supposed to be ¼ inch in diameter, but if measured it will be found to be nearly ⅛ of an inch larger.

The threads on pipes and pipe fittings are also made to standard, and stock taps and dies made for the various sizes of pipe. These taps and dies are spoken of, or described, by stating the size of the pipe for which they are intended. For example, a ¼-inch pipe-tap is a tap of the proper size, shape, and number of threads per inch to cut the thread in a hole to receive a ¼-inch pipe. Threaded holes are made tapering for pipes, the standard taper being ¾ inch per foot, that is, the diameter of the holes decreases at the rate of ¾ inch per foot. In representing a hole which is threaded with a pipe tap the hole is drawn of a diameter at its larger end about equal to the outside diameter of the pipe which is to be screwed into it, and is drawn tapering. It is well to make the taper considerably greater than the actual taper, so that the person looking at the drawing may see at a glance that the hole is for a pipe.

The thread is indicated in one of the conventional ways previously explained, but the number of threads per inch and the diameter of the hole need not be given; instead, a note is made that the hole is tapped for a certain size pipe. Fig. 29 illustrates this.

The following tables of standards for wrought-iron pipe may be found convenient:

STANDARD SIZES OF WROUGHT IRON PIPE

Nominal Size.	⅛	¼	⅜	½	¾	1	1¼	1½	2
Actual Inside Diameter.	.27	.364	.494	.623	.824	1.048	1.38	1.611	2.067
Outside Diameter.	.405	.54	.675	.84	1.05	1.315	1.66	1.90	2.375
Nominal Size.	2½	3	3½	4	4½	5	6	7	8
Actual Inside Diameter.	2.468	3.067	3.548	4.026	4.508	5.045	6.065	7.023	7.982
Outside Diameter:	2.875	3.50	4.00	4.50	5.00	5.563	6.625	7.625	8.625

STANDARD THREADS FOR WROUGHT IRON PIPE

Nominal Size of Pipe.	⅛	¼	⅜	½	¾	1	1¼	1½	2
Threads per Inch.	27	18	18	14	14	11½	11½	11½	11½
Nominal Size of Pipe.	2½	3	3½	4	4½	5	5½	7	8
Threads per Inch.	8	8	8	8	8	8	8	8	8

Scale Drawings. When the object which is to be drawn is not so large but that it can be easily actual size, or full size as it is called, on a sheet of paper which is of convenient dimensions, it is well, usually, to draw the piece full size. In many cases, however, the machine, or the building, or whatever is to be drawn, is so large that it would be impossible to draw it full size. Then the drawing is made to some reduced scale, that is, all the dimensions are drawn smaller than the actual dimensions of the object itself; all dimensions being reduced in the same proportion. For example, if a piece is to be drawn half size, the distance from one point to another on the drawing would be one-half what it is on the piece itself; if the drawing is one-fourth size, the distance on the drawing would be one-fourth what it is on the piece itself, and so on. In dimensioning such a drawing the dimension which is written on the drawing is the *actual dimension* of the piece, and not the distance which is measured on the drawing. This fact must be very clearly understood by the student.

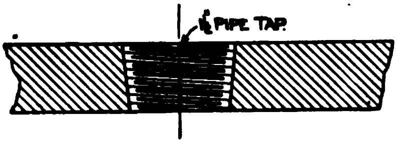
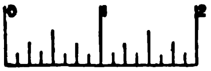
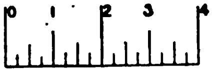
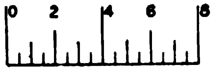


Fig. 29. Conventional Drawing of Thread in a Pipe Fitting.

The common method of reducing all the dimensions in the same proportion is to choose a certain distance and let that distance represent one foot, this distance is then divided into twelve parts and each one of these parts represents an inch; then if half and quarter inches are required these twelfths are subdivided into halves, quarters, etc., until the subdivisions become so small that they cannot be used. We now have a scale which represents the common foot rule with its subdivisions into inches and fractions; but our new foot is smaller than the ordinary distance which we call a foot, and of course its subdivisions are proportionately smaller. When we make a measurement on the drawing we make it with our reduced foot rule and when we make a measurement on the machine itself we make it with the common foot rule.

Draftsmen's scales can be bought which have different distances thus divided, so that if the draftsman wishes to draw a piece one-fourth size he looks over his scale until he finds a distance of three inches (which is of course one-fourth of a foot) divided as explained above, and he uses this to measure with on his drawing. His drawing would then be made to a scale of three inches to the foot. In the same way, if he wishes to make his drawing one-twelfth size he finds on his scale one inch divided into twelfths and fractions of twelfths and uses this as his standard of measurement; if he wishes to make his drawing one forty-eighth size he uses a quarter inch with its subdivisions. Sometimes if the piece to be drawn is very small, the drawing is made to an enlarged scale, such as twice size, three times size, etc.

The mistake of choosing the wrong distance to use on a scale is often made. For example, if he wishes to draw a piece ¼ size, he may look over his scale for a place marked ¼, and use this for his standard for ¼ size, which is wrong. The figure on the scale indicates the distance which is divided up to represent one foot, so that the part of the scale which has ¼ marked on it means that ¼ of an inch is divided up into twelfths, or in other words, if a drawing is made according to that scale it will be $\frac{1}{48}$th size.

The best scales for shop drawings in the United States are those readily derived from the common foot rule, such as full size, 6 inches = 1 foot, 3 inches = 1 foot, 1½ inches = 1 foot. These are the most common, most easily read from an ordinary scale, and one of these can usually be adopted. The student should learn to read these from an ordinary scale without being confined to a special graduation. To do this it is not necessary to divide each dimension by 2, 4, and 8 to get half size, quarter size, or eighth size, and then lay down the result. For half size, or 6 inches = 1 foot, ½ inch on an ordinary rule represents 1 inch. Hence, each half inch may be read as 1 inch, and its subdivisions accordingly, thus:  For 3 inches = 1 foot, or quarter size, ¼ inch represents 1 inch, and looks thus:  For 1½ inches = 1 foot, or eighth size, ⅛ inch represents 1 inch, and looks thus:  It is very easy to get accustomed to this, and it saves much time and trouble hunting up a special scale every time.

The other allowable scales, less common, but sometimes necessary on large work, are 1 inch = 1 foot, ¾ inch = 1 foot, ½ inch = 1 foot, ⅜ inch = 1 foot, ¼ inch = 1 foot, and ⅛ inch = 1 foot. To use these scales conveniently, special graduation is desirable.

Every drawing should have the scale to which it is made plainly marked upon it, as a part of, or adjacent to the title.

Fig. 30 shows a detail which has been reduced in making the cut so that its dimensions are on a scale of 3 inches per foot, or quarter size. By applying his scale to this cut and comparing the readings of his scale with the dimensions as given, the student will gain a clear understanding of a reduced scale drawing. None of the other cuts in this book are reduced for the purpose of scaling, hence have no even relation to their dimensions as given.

GENERAL SYSTEM FOR SHOP DRAWINGS

The principles of detail drawings having been thoroughly discussed in the preceding pages, the general system to be followed in preparing shop drawings for the workman's use will now be outlined.

Sketches. As previously stated, for a new machine, the original sketches will be supplied by the designer, and it is the duty of the detail draftsman to read and interpret them; or the designer may furnish a rough general layout to scale, from which the detail draftsman must pick out the details, scaling off the dimensions.

Oftentimes certain details of an existing machine have to be copied, in which case the sketches will have to be made by the detail draftsman himself, from the machine. Proficiency in the art of making sketches is a very valuable and necessary acquisition for any draftsman. Accuracy, completeness, clearness, and rapidity in making, are the principal requirements for a good sketch.

The sketches should be made so clear, that if they are laid aside for a long time they can be readily understood without depending at all upon memory. There is a strong tendency for the beginner to make his sketches hurriedly, thinking that when he comes to finish his drawing he can supply the details from memory. This is a bad plan and will lead to many mistakes. The sketches must be so clear and complete that anyone can read them who has never seen the machine. No attempt need be made to draw them to scale,

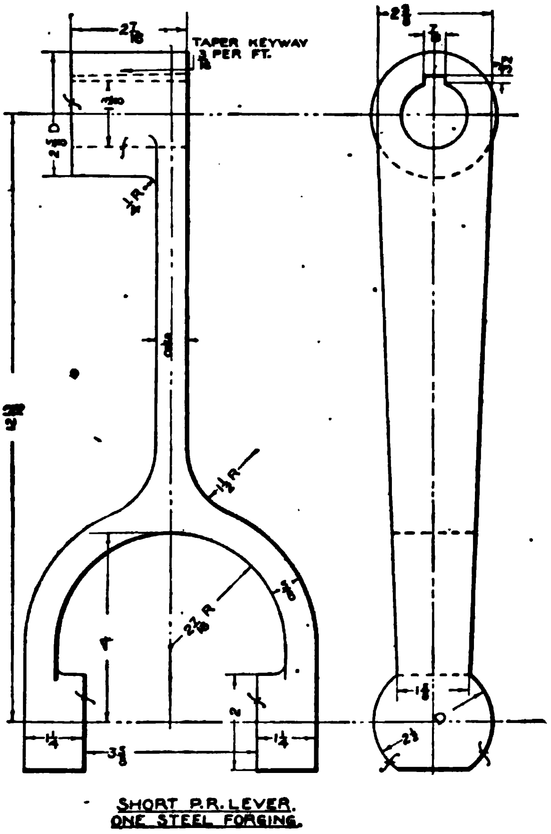


Fig. 30. Reduced Scale Drawing. Actual Scale, 3"=1'.

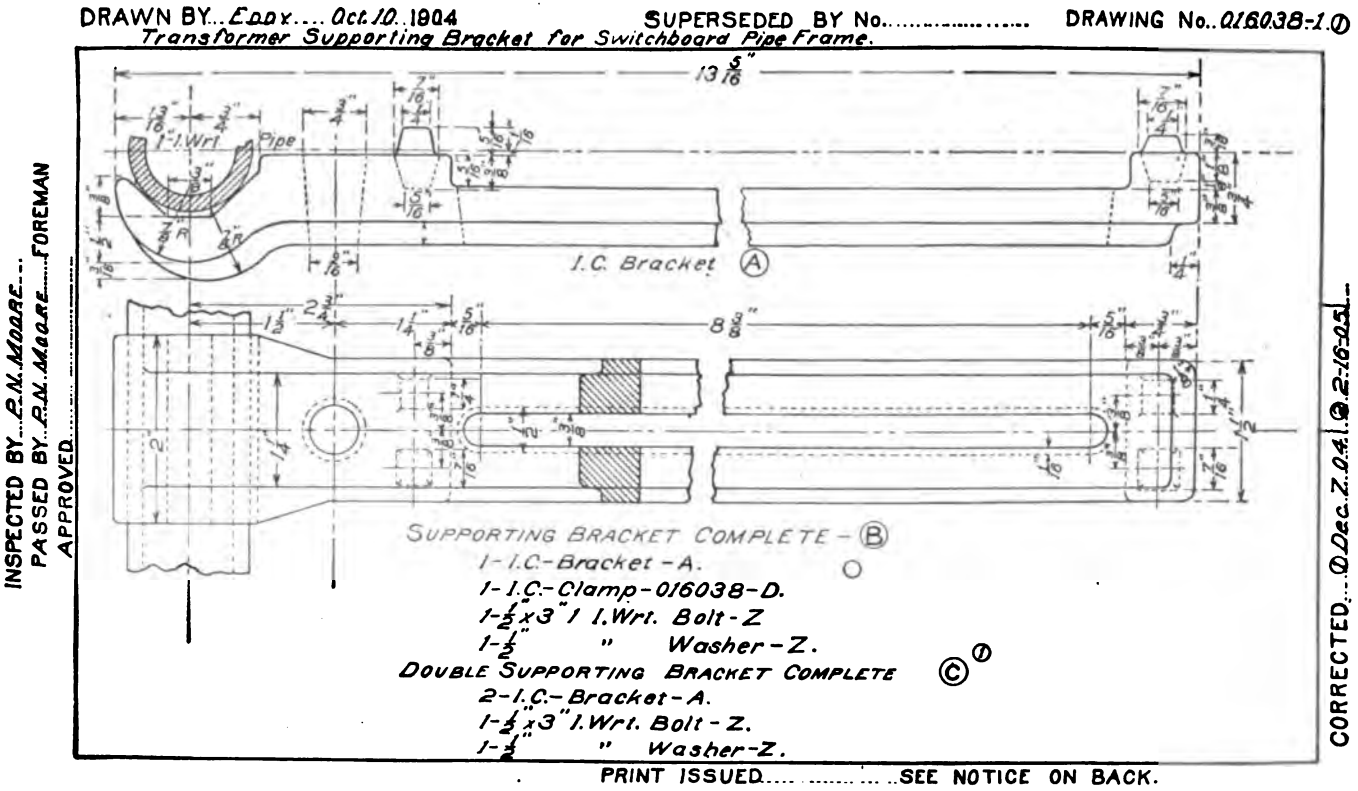
but *all dimensions*, carefully measured from the machine, should be placed on the sketch.

Pencil Drawing. After the sketches are made, the next step is the making of the pencil drawing from the sketches, accurately to scale. The size of the plate on which the drawing is to be made is usually fixed by some standard. Where many drawings are made and kept in an office, it is desirable to keep the plates of uniform size, as far as possible. It is good practice to have two or three standard sizes of plates, one for small, one for medium, and one for large drawings.

Assuming, then, that we have our paper tacked on the drawing board and the plate laid out, the next step will be to arrange the drawings of the various pieces on the plate so that there will be room for all and so that they may be properly placed with relation to each other. It may happen that there will not be room on one plate for all the pieces, but that two or more plates will be required. When the parts must be thus arranged on different plates, an effort should be made to keep on the same plate those parts which belong together. For example, if we were drawing a lathe, the details of the parts of the head stock might form one plate, the apron another, and so on.

In locating the various pieces on a plate, they should be placed as nearly as possible in the same relative position to each other that they bear in the machine, except that they are separated. For example: if a nut belongs on the end of a screw, it is desirable to draw it on the same center line with the screw and at the end where it belongs. If a piece is vertical in the machine, it should be vertical on the plate, and if horizontal in the machine, it should be horizontal on the plate.

The approximate location of the pieces on the plate may be easily decided by taking a small sheet of paper of about the same proportion as the plate, but perhaps one-fourth or one-half size, and sketching on it roughly the outline of the various pieces. The arranging of the plate should not be allowed to take much time, but should be done as rapidly as possible. After the location of each view of each piece is determined, the pencil drawing should begin (to scale) with one of the principal pieces. In almost all cases a center line is first drawn. It is better to carry along all


DRAWN BY...Eddy....Oct.10.1904
SUPERSEDED BY No..................
DRAWING No..016038-1.①
Transformer Supporting Bracket for Switchboard Pipe Frame.
13 5/16"
1" I.Wrt. Pipe
I.C. Bracket Ⓐ
8 3/8"
SUPPORTING BRACKET COMPLETE – Ⓑ
1-I.C.-Bracket-A.
1-I.C.-Clamp-016038-D.
1-1/2"x3" I.Wrt. Bolt-Z
1-1/2" " Washer-Z.
DOUBLE SUPPORTING BRACKET COMPLETE Ⓒ①
2-I.C.-Bracket-A.
1-1/2"x3" I.Wrt. Bolt-Z.
1-1/2" " Washer-Z.
PRINT ISSUED.................SEE NOTICE ON BACK.
INSPECTED BY...P.N.Moore...
PASSED BY..P.N.Moore.....FOREMAN
APPROVED.....................
CORRECTED...① Dec.2.04.|② 2-16-05.

TYPICAL SHOP DRAWING MADE IN THE DRAFTING ROOM OF THE STANLEY ELECTRIC MFG. CO.

the views of a piece at once, instead of completing one view at a time. The piece started should have all its views finished and completely dimensioned before another piece is begun; exceptions to this are sometimes necessary for special reasons. The lines should be drawn accurately, but no attempt need be made to obtain finish; thus, in order to save time, the lines may be run past the point where they should properly stop, etc. Nothing should be omitted, however.

Tracing. Having finished the pencil drawing, the next step is the inking. In some offices the pencil drawing is made on a thin, tough paper, called bond paper, and the inking is done over the pencil drawing, in the manner with which the student is already familiar. It is more common to do the inking on thin, transparent cloth, called tracing cloth, which is prepared for the purpose. This tracing cloth is made of various kinds, the kind in ordinary use being what is known as "dull back," that is, one side is finished and the other side is left dull. Either side may be used to draw upon, but most draftsmen prefer the dull side.

The tracing cloth is stretched smoothly over the pencil drawing and a little powdered chalk rubbed over it with a dry cloth, to remove the slight amount of grease or oil from the surface and make it take the ink better. The dust must be carefully brushed or wiped off with a soft cloth, after the rubbing, or it will interfere with the inking.

The drawing is then made in ink on the tracing cloth, after the same general rules as for inking on paper, but care must be taken to draw the ink lines exactly over the pencil lines on the paper underneath, which should be heavy enough to be easily seen through the tracing cloth. The ink lines should be firm and heavy to assure good blue prints. In tracing, it is better to complete one view at a time, because if parts of several views are traced and the drawing left for a day or two, the cloth is liable to stretch and warp so that it will be difficult to complete the views and make the new lines fit those already drawn and at the same time conform to the pencil lines underneath. For this reason it is well, when possible, to complete a view before leaving the drawing for any length of time, although of course on views in which there is a good deal of work this cannot always be done. In this case the draftsman must manipulate his tracing cloth and instruments to make the lines fit as best he can. A skillful draftsman will have no trouble from this source, but the beginner may at first find difficulty.

Inking on tracing cloth will be found by the beginner to be quite different from inking on the paper to which he has been accustomed, and he will doubtless make many blots and become discouraged with his first attempt to make a tracing. After a little practice, however, he will find that the tracing cloth is very satisfactory and that a good drawing can be made on it quite as easily as on paper.

The necessity for making erasures should be avoided, as far as possible, but when an erasure must be made a good ink rubber or typewriter eraser may be used. If the erased line is to have ink placed on it, such as a line crossing, it is better to use a soft rubber eraser. All moisture should be kept from the cloth.

Blue Printing. The tracing, of course, cannot be sent into the shop for the workmen to use, as it would soon become soiled and in time destroyed, so that it is necessary to have some cheap and rapid means of making copies from it. These copies are made by the process of blue printing, in which the tracing is used in a manner similar to the use made of a negative in photography.

Almost all drafting rooms have a frame for the purpose of making blue prints. These frames are made in many styles, some simple, some elaborate. A simple and efficient form is a flat surface usually of wood, covered with padding of soft material, such as felting. To this is hinged the cover, which consists of a frame similar to a picture frame, in which is set a piece of clear glass. The whole is either mounted on a track or on some sort of a swinging arm, so that it may readily be run in and out of a window.

The print is made on paper prepared for the purpose by having one of its surfaces coated with chemicals which are sensitive to sunlight. This coated paper, or blue-print paper, as it is called, is laid on the padded surface of the frame with its coated side uppermost; the tracing laid over it right side up, and the glass pressed down firmly and fastened in place. Springs are frequently used to keep the paper, tracing, etc., against the glass. With some frames it is more convenient to turn them over and remove the backs. In such cases the tracing is laid against the glass, face down; the coated paper is then placed on it with the coated side against the tracing cloth.

The sun is allowed to shine upon the drawing for a few minutes, then the blue-print paper is taken out and thoroughly washed in clean water for several minutes and hung up to dry. If the paper has been recently prepared and the exposure properly timed, the coated surface of the paper will now be of a clear, deep blue color, except where it was covered by the ink lines, where it will be perfectly white.

The action has been this: Before the paper was exposed to the light the coating was of a pale yellow color, and if it had then been put in water the coating would have all washed off, leaving the paper white. In other words, before being exposed to the sunlight the coating was soluble. The light penetrated the transparent tracing cloth and acted upon the chemicals of the coating, changing their nature so that they became insoluble; that is, when put in water, the coating, instead of being washed off, merely turned blue. The light could not penetrate the ink with which the lines, figures, etc., were drawn, consequently the coating under these was not acted upon and it washed off when put in water, leaving a white copy of the ink drawing on a blue background. If running water cannot be used, the paper must be washed in a sufficient number of changes until the water is clear. It is a good plan to arrange a tank having an overflow, so that the water may remain at a depth of about 3 or 4 inches.

The length of time to which a print should be exposed to the light depends upon the quality and freshness of the paper, the chemicals used and the brightness of the light. Some paper is prepared so that an exposure of one minute, or even less, in bright sunlight, will give a good print, and the time ranges from this to twenty minutes or more, according to the proportions of the various chemicals in the coating. If the full strength of the sunlight does not strike the paper, as, for instance, if clouds partly cover the sun, the time of exposure must be lengthened.

Blue print paper should not be exposed to bright actinic light except during the process of printing. It is not, however, so sensitive as the ordinary dry plate and may be handled in a subdued light if the exposure is very brief. When not in use the paper must be kept in a dry, dark place and should be hermetically sealed.

A more modern type of blue print machine is the electric machine, usually arranged in the form of a vertical cylinder of glass, around which is placed the sensitized paper and the tracing, and along the longitudinal axis of which travels an electric arc lamp at a uniform speed. This speed is so adjusted that the right exposure is given to the sensitized paper. There are several other types of electric blue printing machines, all based upon the same idea of uniform exposure of the sensitized paper to the rays of one or more arc lamps. These machines are a positive necessity to modern drawing offices, because of the uncertainty of sunlight, and, therefore, limited capacity for turning out prints. With the electric machines there is no limit, and they may be run to their full capacity 24 hours a day if desired.

FORMULA FOR BLUE-PRINT SOLUTION

Dissolve thoroughly and filter.

A.	Red Prussiate of potash	2½ ounces,
	Water	1 pint.
B.	Ammonio-Citrate of iron	4 ounces,
	Water	1 pint.

Use equal parts of A and B.

FORMULA FOR BLACK PRINTS

Negatives. White lines on blue ground; prepare the paper with

Ammonio-Citrate of iron	40 grains,
Water	1 ounce.

After printing wash in water.

Positives. Black lines on white ground; prepare the paper with:

Iron perchloride	616 grains,
Oxalic Acid	308 grains,
Water	14 ounces.

Develop in	Gallic Acid	1 ounce,
	Citric Acid	1 ounce,
	Alum	8 ounces.

Use 1¼ ounces of developer to one gallon of water. Paper is fully exposed when it has changed from yellow to white.

General Drawing. We have followed through the process of making a detail drawing, from the sketches to the blue print ready for the workmen. Such a detail drawing or set of drawings shows the form and size of each piece, but does not show how the pieces go together and gives no idea of the machine as a whole. Consequently, a *general drawing* or *assembled drawing* must be made, which will show these things. Usually two or more views are necessary, the number depending upon the complexity of the machine. Very often a cross-section through some part of the machine, chosen so as to give the best general idea with the least amount of work, will make the drawing clearer.

The number of dimensions required on an assembled drawing depends largely upon the kind of machine. It is usually best to give the important over-all dimensions and the distance between the principal center lines. Care must be taken that the over-all dimensions agree with the sum of the dimensions of the various details. For example, suppose three pieces are bolted together, the thickness of the pieces according to the detail drawing, being one inch, two inches, and five and one-half inches respectively; the sum of these three dimensions is eight and one-half inches and the dimensions from outside to outside on the assembled drawing, if given at all, must agree with this. These over-all dimensions serve as a check and relieve the mechanic of the necessity of adding fractions.

ILLUSTRATIVE DRAWINGS

The remaining pages of this Instruction Paper are illustrative of common practice in making working drawings, which it would be tedious and difficult to formulate as rules to guide the student. By a careful study of the illustrations, all of which are practical working drawings of a variety of pieces, and a close following of the description, more can be learned as to making a drawing than by adhering to a multitude of rules. After a study of the preceding pages, expounding the principles involved, the student will find most of his further questions answered by reference to Figs. 31 to 50 inclusive. While he may find that the methods of lines and dimensions shown in these figures are in some ways different from those he may see or hear of from other quarters, he should remember that the language of drawing differs widely in usage. The result desired, however, is always the same, namely, *complete instructions to the workman.* This is the sole test of a good drawing, however made, and the student's aim should always be to satisfy this requirement.

In making working drawings of a machine the detail draftsman must secure the several dimensions of the parts, either by measuring the general layout which has been made by the designing draftsman, or he may be given a rough sketch of the part, which he is to develop into an exact working shop drawing. Sometimes he may himself be called upon to go into the shop and measure up the parts of an existing machine, making his own sketches, and then detail drawings from them. Such sketches, while of the roughest kind, must be accurately and completely dimensioned at the time the sketch is made, as it is not always convenient or possible to make subsequent trips to the machine to fill in lacking dimensions on the sketch. The making of satisfactory sketches is not as easy as would at first appear, but is quite an art in itself, acquired by systematic action and experience.

An illustration of a sketch of this kind is shown in Fig. 31, the subject being a crank, keyed and clamped to a shaft. The first thing to do is to roughly sketch the piece with sufficient care to enable the dimensions to be put on. (A soft or medium pencil is the best for such purpose and any scrap of paper or a sketching pad will suffice.) Each portion of the piece should then be *separately considered* and carefully gone over to see that it is not only properly located but that it has the three dimensions, *length, breadth* and *thickness* properly noted.

Thus in Fig. 31, the large hub should first be located by giving its distance $10\frac{1}{8}''$ from the center of the smaller hub, then its diameter $5\frac{3}{4}''$ should be given, the distance between its faces $3\frac{1}{2}''$, and $6''$ radius between them; next, the diameter of the hole to receive the shaft $2\frac{3}{16}''$, after which it is noted that it has a key-way, the dimensions of which are necessary. Then the boss for the clamping bolt should be located by the figure $2\frac{3}{8}''$ to its center, its location in the other direction being on the center line of the arm. This boss has a diameter of $2\frac{3}{8}''$ and a length of $2\frac{1}{4}''$ each side of the center to the finished surface, the depth of the counter-bore being $\frac{1}{16}''$, and the width of the slot to the bore $\frac{1}{4}''$, all of which dimensions should be carefully put on. The boss has a hole in it tapped at one end for a $1''$ bolt and drilled at the other end $1\frac{1}{32}''$. This completes

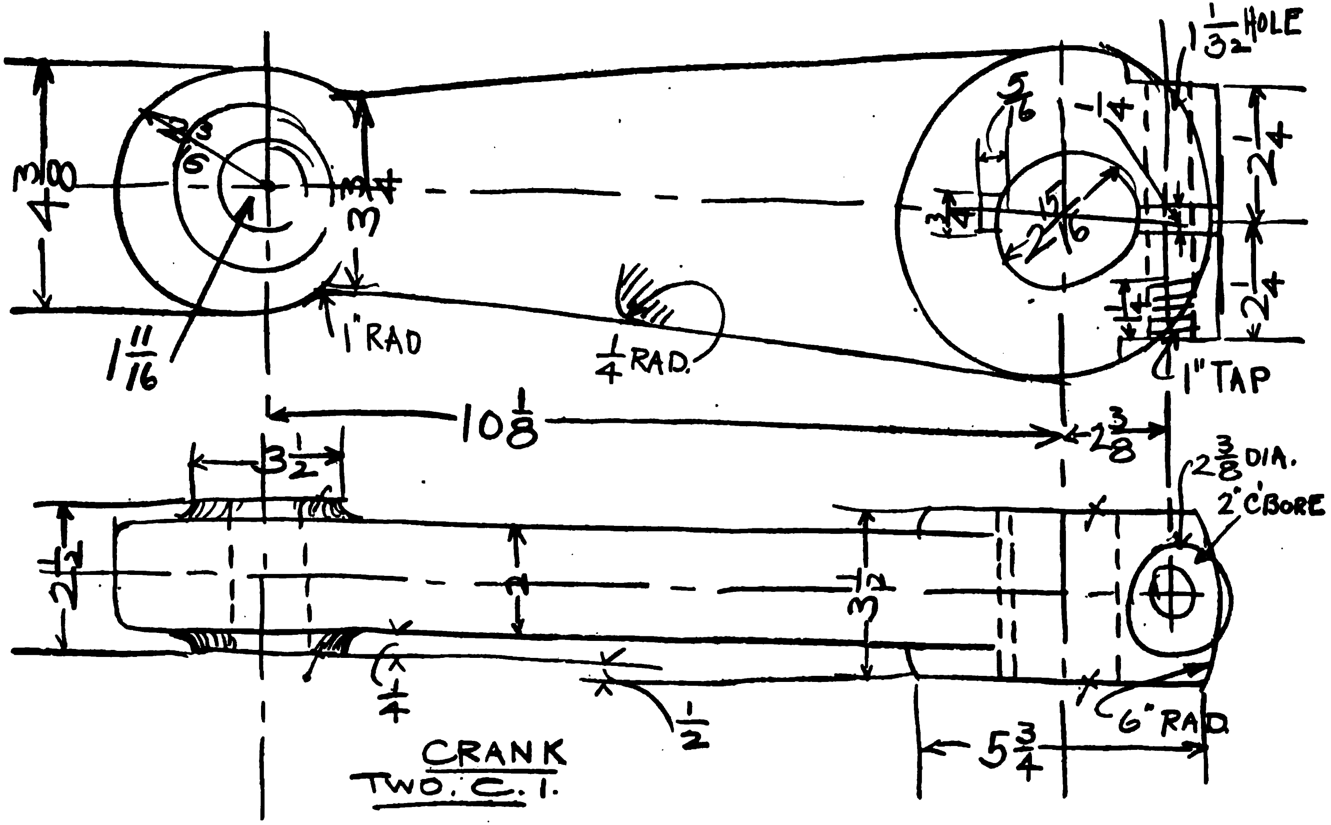


Fig. 81. Preliminary Sketch of Crank.

the figuring of the large hub and we can proceed to dimension the other end of the arm. This has a diameter of $4\frac{3}{8}''$, the thickness of the arm being $2''$, and there are facing pads on either side $\frac{1}{4}''$ high, bringing the total distance from face to face $2\frac{1}{2}''$; in order to show positively that these portions are central with the faces of the large hub, the figure $\frac{1}{2}''$ is put at one side; the diameter of these facing pads is $3\frac{1}{2}''$, and the hole through the head of the crank is $1\frac{1}{16}''$ diameter. Having put on the above figures we now have to provide a connection between the head of the crank and the hub, and it therefore becomes necessary to give figures for the size of the arm; the thickness of the arm has been already given as $2''$, and the width being the same as the diameter of the hub, the side lines are simply drawn tangent to the same; at the smaller end the width may be conveniently given along a line tangent to the facing pad as $3\frac{3}{4}''$. The arm is filleted into the hub by $1''$ radius. The only thing now uncertain is whether the corners of the arm are sharp or rounded, and this is shown by the little section of the corner giving $\frac{1}{4}''$ radius.

The above description is tedious and the dimensions can probably be put on more quickly than the discussion of them can be read, but it should be especially noted that the systematic method has been followed of taking each part of the piece separately and dimensioning it before taking up any other part. While this is not always entirely possible to do in complicated pieces, yet it is absolutely necessary that in general this principle be always followed; otherwise, it is impossible to be sure that all the dimensions are on.

The description above also applies to the dimensioning of the piece after it is drawn in detail, this being represented in Fig. 32.

Referring to this figure, the bold character of the drawing should be noted, the solid lines being strong and of absolutely different character from the center or dimension lines. There is no uncertainty about the direction or termination of the lines; the figures are bold, plainly made, and absolutely clear; there can be no possible excuse for the workman to read any of the lines or dimensions wrongly. In other words, the drawing satisfies the definition of a working drawing, as previously given, in that *it conveys absolutely* *definite instructions to the workman, expressed in the simplest and most straightforward way.*

On most machine parts a portion only of the surfaces are finished; and these are usually indicated, as previously noted, by the small letter "f" placed across the line representing the surface; this indicates to the pattern maker that he is to allow extra stock on the pattern, so that when the rough casting is made there will be sufficient metal to enable the finishing cut to be taken to the proper dimension. These finished surfaces are the most important surfaces of the piece, to which all the other parts have to be related. In order that the several parts of the machine may properly go together, it is necessary for the draftsman, in putting on the figures, to start from some one finished surface, and so arrange the figures that the machinist can readily work from one finished surface to another. In Fig. 32 the dimensions of the rough parts as given may not be exactly maintained in the casting, but the distances between the finished surfaces

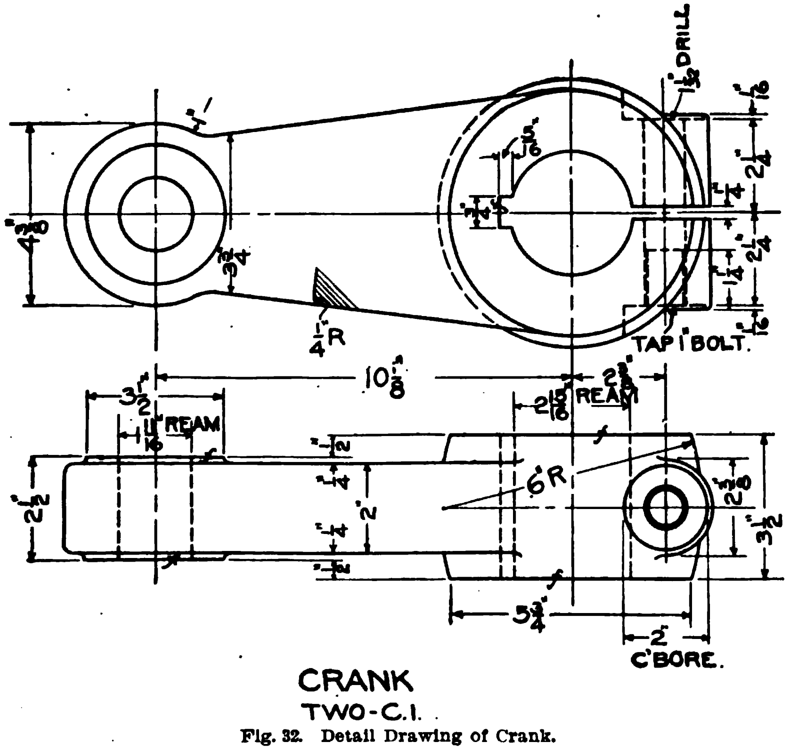


Fig. 32. Detail Drawing of Crank.

must be exactly secured. The method of figuring a key-way is illustrated in this figure, and it should be carefully noted that the depth of the key-way ($\frac{5}{16}$ inch) is given from the corner where the side of the key-way intersects the bore; this is because the depth of the key-way is readily measured by scale from this point.

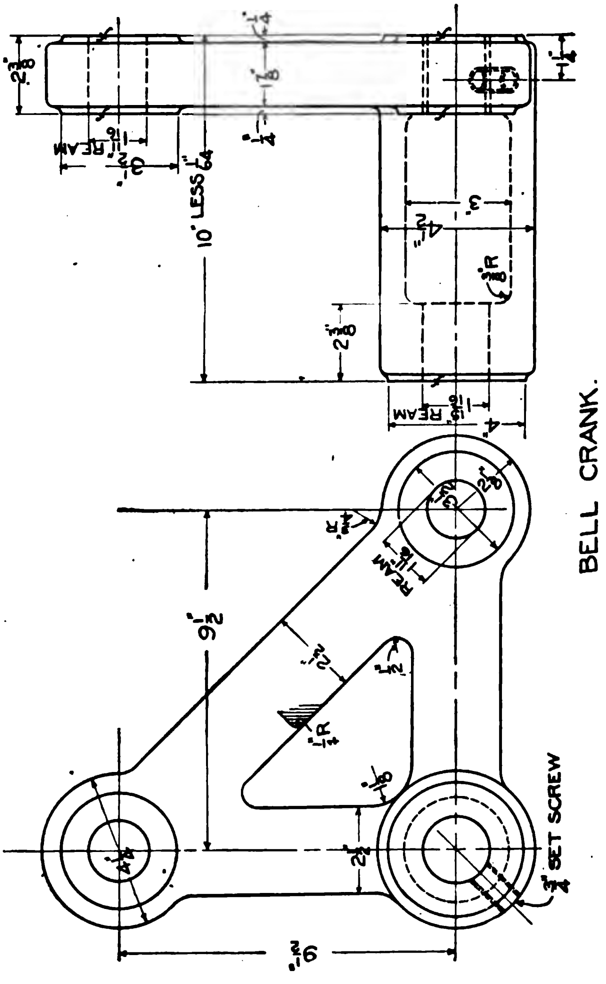


Fig. 33. Method of Showing Dimensions on Detail Drawing.

The thread for the 1-inch bolt is indicated in this case by a double line, the inside line representing the bottom, the outside representing the top of the thread, while the lines of the helix are entirely omitted. This is not as common a method of representing a thread as the conventional method previously described.

Fig. 33 shows a bell crank fastened to its shaft by means of a set screw, the same general features being noted in this as in the preceding figure. A further point is the method of expressing the distance between the faces of the principal hub and the smaller hub, "10 inches less $\frac{1}{64}$ inch." This method of stating a dimension is quite common among certain manufacturers, as it saves giving odd dimensions and conveys more quickly to the workman's mind what the dimension is. The other method of stating this would be "$9\frac{63}{64}$ inches," which is obviously a somewhat cumbersome and odd dimension; it is easier for the machinist to read 10 inches on his scale and finish the distance $\frac{1}{64}$ inch less, than it is to use the actual figure. This point is an instance in which the instructions furnished by the drawing to the workman are simplified for his benefit.

Another point worth noting in this figure is that circles are dimensioned by giving the diameters in preference to the radii; this is for the benefit of the pattern maker and the machinist, who always use calipers for measuring these parts. When the radii are given, the workman is forced to multiply the radius by two in order to secure the dimension for his calipers; and it is always better to remove the chance of error through the workman in the shop making mathematical calculations. A detailed drawing should be so completely dimensioned that there will be no occasion for the workman to make any calculations himself; for, even if he is competent to do it, the responsibility for the correctness of the figures should be on the draftsman. In practically all shops the workmen are not allowed to scale the drawings in case dimensions are lacking, but are required to go to the drafting room and have further dimensions put on as required. A good, detail drawing should require no such additions.

Fig. 34 shows the detail of a rocker-arm and pin, such as are frequently found in connection with the valve gear of a steam engine. This is a case where it is just as clear to detail two pieces together as to separate them entirely, the rocker-arm pin being shown in position in the large hub and dimensioned in that position. This is not only simpler, but it gives the added information to the workman of just how the pin goes in the arm, and enables him to make his fits accordingly.

This principle of detailing several pieces together may, however, be carried to the point where the drawing becomes confused and complicated; then it is best to separate the detail of the parts. There are often special reasons requiring the detailing of parts of one kind on one sheet and one on another; for example, some shops detail forgings on one sheet, castings on another, parts to be made on the screw machine on another, and turret lathe work on another, etc., such arrangements being dependent upon the particular shop organization to which they apply.

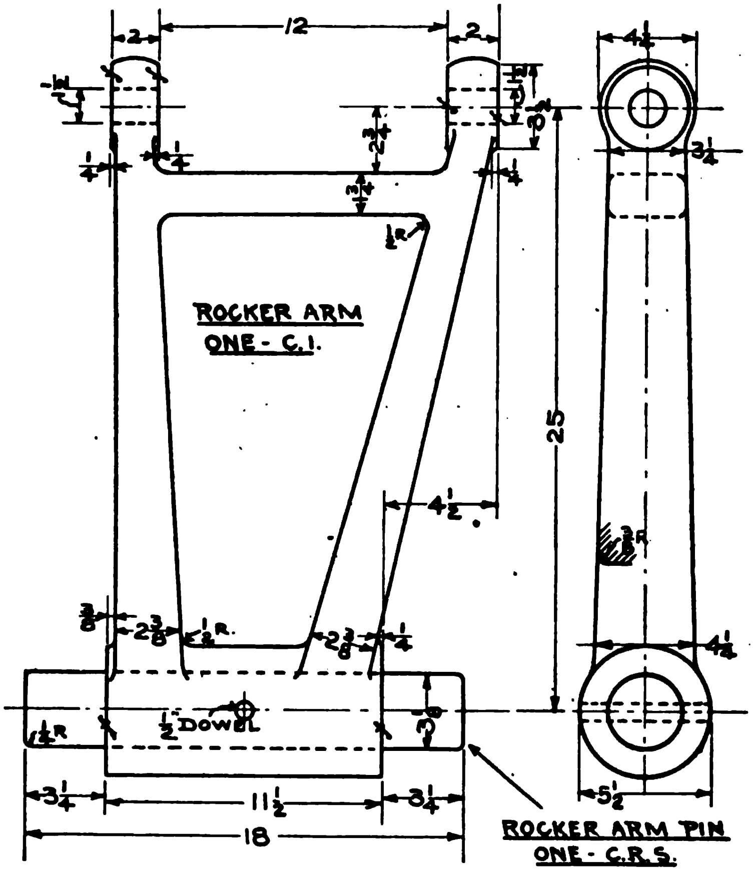


Fig. 34. Detail of Rocker Arm and Pin.

Attention is called in this figure to the fact that inch marks are not placed after any of the dimensions. In cases where there can be no misunderstanding of what is meant by the figures, as in the present case, the inch marks are wholly unnecessary and may be left off. This is the case with most drawings; and in most of the illustrations of this book, the inch marks are not used, except where dimensions are given in feet and inches, when they are obviously necessary. It is a customary practice, however, to print these inch marks after all figures; and there is no objection to doing so, except for the fact that it is usually an unnecessary waste of time and labor.

Fig. 35 shows a link stud, also used in connection with the valve gear of an engine. On such pieces as this it is usually considered that they are finished all over unless otherwise mentioned. It is always desirable, in finished pieces of this character, to give the length *over all* of the pieces, in order that the workman may quickly determine how much stock to order from the stock room without having to add up the figures between the various shoulders of the piece. The head of this stud is a hexagon, and the usual method of calling for the hexagon is given in the note, "2 across flats." The reason the distance across flats is specified is to enable the workman to see at once what the necessary width of the jaws of the wrench must be in order to fit the head of the stud.

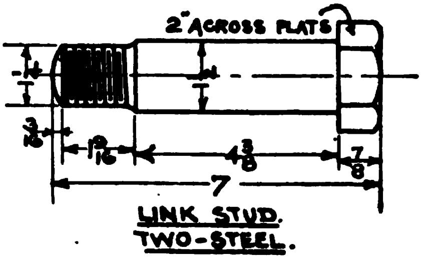
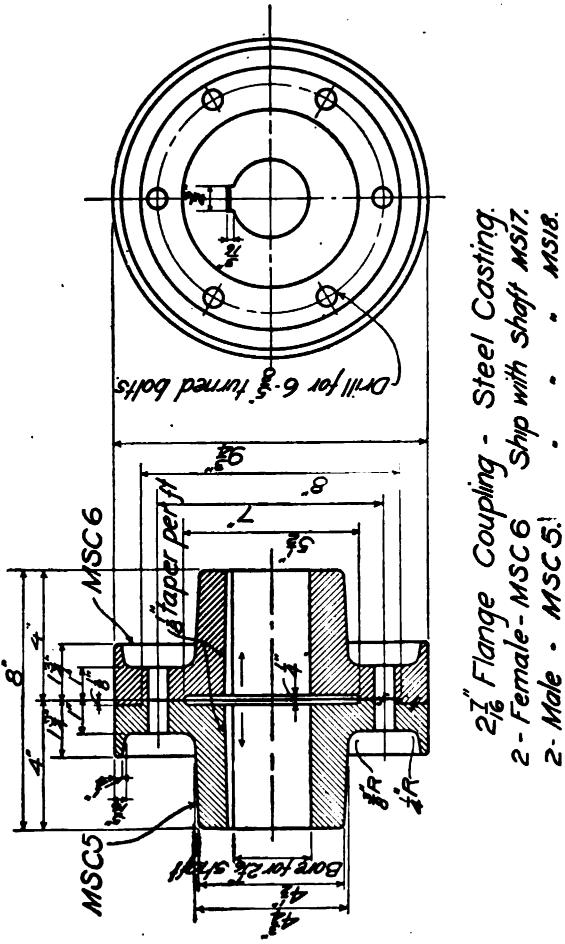


Fig. 35. Detail Drawing of Link Stud.

Fig. 36 shows a flange coupling such as would be used to connect up a line shaft in a shop, or any heavy machinery shafting. The detail construction of this is most clearly shown by making the drawing in cross-section. This drawing is a good illustration of the placing of the dimensions entirely outside the lines of the drawing, thus enabling any changes to be made in the figures without in any way obliterating the drawing. It also keeps both drawing and figures definite and clear, avoiding any possibility of confusion. Each half of the coupling is fastened to its shaft by a tapered key; and in order that it may be clearly understood which way the top of the key-way is tapered, the arrows, shown in the cross-sectional view, indicate the direction in which the key is driven home.

On this drawing are indicated the pattern numbers. They would be equally necessary on all other castings illustrated in this book; but for purposes of simplicity they have generally been omitted. Pattern numbers are necessary, not only that the patterns may be filed away systematically, and readily found when wanted, but also that the necessary orders for the castings may be written, and that the pieces may be identified on the drawings. For wrought-iron and steel pieces which have no patterns, certain letters or numbers are given them, such that the identification is as complete as with



2 7/16" Flange Coupling - Steel Casting.
2 - Female - MSC6 Ship with shaft MS17.
2 - Male - MSC5! " " MS18.
Drill for 6-5/8 turned bolts
Bore for 2 7/16 shaft
taper per ft
MSC6
MSC5
8"
4"
4"
8"
7"

Fig. 36. Detail Drawing of a Coupling, Using a Cross-sectional View.

castings. Castings, whenever possible, carry on their surface the pattern numbers in raised figures, and when received in the shop or field can thus be identified for assembling in the machine. Wrought-iron and steel pieces, which cannot have such figures raised upon them, are usually marked with painted letters and figures, to correspond with piece marks called for on the drawing, thus enabling them to be properly identified at the shop and in the field. There are many different systems in vogue for this numbering, dependent upon the particular requirements of the shop organization to which it applies.

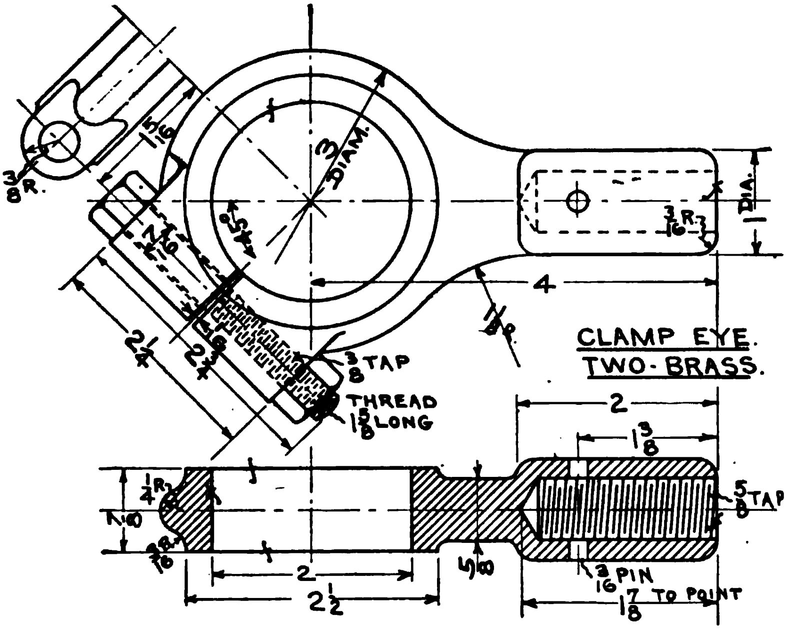


Fig. 37. Detail Drawing with a Projection Thrown Off at an Angle to the Principal Center Lines.

Fig. 37 shows a piece designed to receive a threaded rod at one end and rigidly clamp to a shaft by means of a bolt at the other. This detail, simple as it appears, is awkward to make, on account of the bolt boss being at an angle with the principal center lines. The lower view is a cross-section, because, if the ordinary elevation were shown, it would have a series of ellipses showing the bolt boss projections. It is always desirable to avoid oblique projections of circular shapes on account of the difficulty of drawing same; moreover, the ellipses produced do not show the construction as plainly as a straight projection. The method adopted in such cases is to throw off a straight projection at the same angle as the part in question makes with the principal center lines. In the present instance it is necessary to show that the boss for the bolt is ⅜-inch radius, and that the boss is centrally located with the hub, which is readily done by the straight projection thrown off. In the cross-sectional view, the lines representing the thread appear to the eye sloped in the wrong direction, or as though the thread were left-hand. A moment's thought, however, will convince the student that, since the section taken is through the middle of the hole, we are merely looking at the back side of the hole, and that the threads of a right-handed screw on the back side must necessarily slope in the direction as shown. In the case of the thread on

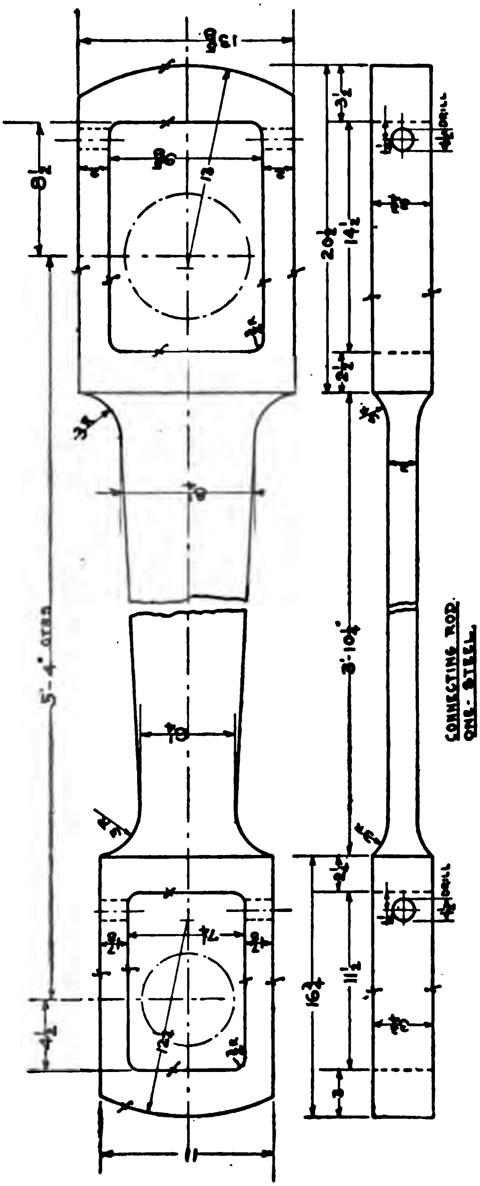


Fig. 38. Detail of a Connecting Rod.

the bolt for the clamping hub, shown dotted, the lines of the thread appear right-handed to the eye, it being universal practice in the case of dotted threads to show the side only next the eye. If the threads on the back side of the bolt were also shown, they would slope in the other direction, crossing the other lines, and to draw them in would obviously cause confusion.

Fig. 38 shows the connecting rod for a small steam engine. This piece calls for little comment. The outlines of the crank pin and cross-head pin are shown in dot-and-dash circles, and the relation of these centers to the rod is given. This is of some importance in enabling the detail of the boxes, which go in the heads of the connecting rod, to be correctly detailed and checked. It should be carefully noted by the student, that any such information which can be given on the details, without confusing the drawing, to tie up one detail with another, is usually good practice. It not only saves time in the drafting room in checking and general reference work, but it gives the workman a better idea of how the parts are expected to go together, thus fulfilling the general definition of a working drawing as "complete and definite instruction."

Fig. 39 shows a gear with a split hub, the bolts through the hub being for the purpose of tightly clamping same to the shaft. This is an illustration of the conventional method of showing a gear with standard proportions of teeth. If the drawing were made exactly as the gear would look, it would be necessary to spend a large amount of time inking in the outlines of the 72 teeth around the circumference of the gear; instead of doing this, the pitch line of the gear is shown, and circles drawn indicating the top and bottom of the teeth. The pitch diameter and outside diameter are given, and the proper depth of tooth cut to be made by the gear cutter; nothing more is necessary. The information which enables the workman to choose the proper cutter is given in the title; thus, "Cut 72 Teeth—1 Diametral Pitch—Involute." The workman will receive from the tool room a standard involute cutter marked for 72 teeth, 1 diametral pitch. After placing same on the arbor of the gear-cutting machine, he will drop the cutter into the gear blank to the depth called for by the drawing; if this is accurately done, it will give the desired standard thickness and clearance of the teeth, and the automatic indexing of the machine will produce the 72 teeth, accurately spaced. If the gear were a cast gear—in which case the pattern maker would have to lay out the teeth and finish them one by one on the pattern—it would be necessary to draw one

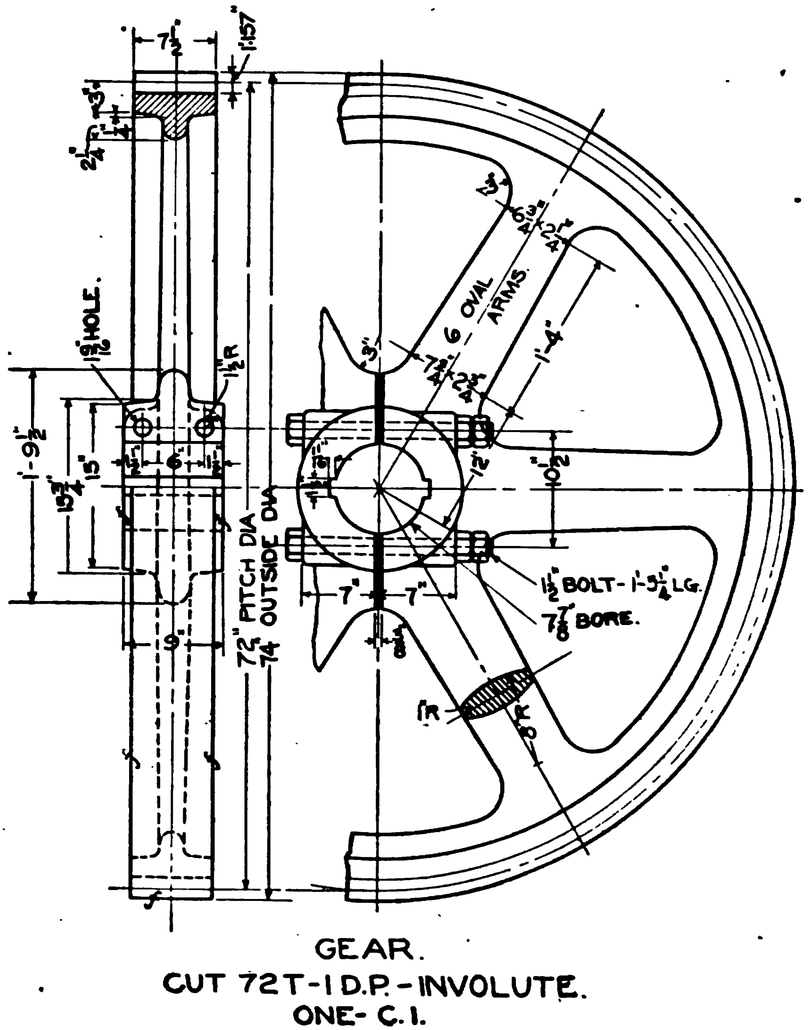


Fig. 39. Detail of Gear with Split Hub.

or two teeth on the circumference of the gear, and give sufficient figures to enable him to lay out the teeth either by his dividers or by a template. In neither case, however, should the detail draftsman take the time to draw the teeth in any great number around the circumference of the gear; this is a very common thing for the young draftsman to do, and nothing shows his greenness more. He should bear in mind that when the proper instruction has been conveyed by his drawing, any more time spent on it is wasteful.

The cross-section of the gear is shown to enable the necessary figures to be placed upon the rim and hub. In the right-hand view, only one-half the complete circle of the gear is shown; nothing would be gained by showing the other half, therefore it would be a waste of time and space on the drawing to make it.

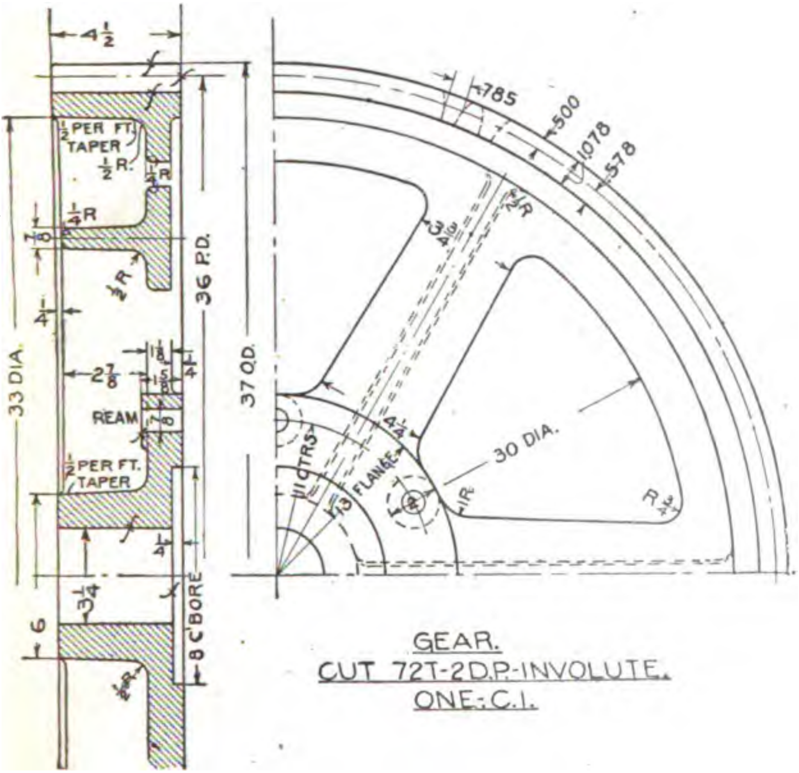


Fig. 40. Detail of Gear with Solid Hub.

Attention is called to the method of indicating the dimensions of the arms, the breadth and thickness of the oval being indicated as follows: "6¾″ × 2¼″;" this is a "short cut" which will usually answer, but it does not actually show the section of the arm. It would be better actually to make a cross-section, as shown in the lower portion of the right-hand view, giving the actual radii for the section, as otherwise the pattern maker might make an arm more or less blunt on the ends than the draftsman intended.

Fig. 40 shows the detail of a spur gear with a T-shape arm. The gear drives through fitted bolts in the flange about the hub.

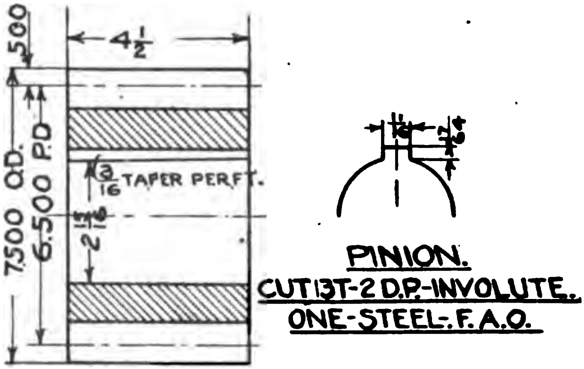


Fig. 41. Detail of Pinion.

Two or three teeth are dotted in, to show their dimensions, which are according to the standard involute system.

Fig. 41 shows the detail of a pinion designed to mesh with the gear in Fig. 40, one view being sufficient.

Fig. 42 shows the detail drawing of a pair of bevel gears. By careful study of this drawing, the student will gain an idea of the dimensions to be shown on a bevel gear.

The gear-cutting machines, on which the teeth of these gears are cut, require the angles as given for the setting of the cutters. It should also be remembered that the casting must be finished by the machinist before the teeth are cut; hence the dimensions for hubs, diameters, etc., must be so put on that they can be conveniently used by the machinist in turning up the gear blank in an ordinary lathe.

Fig. 43 shows the detail of a worm and worm gear. The teeth on the worm gear are twisted; and if it were attempted to show their true projection, it would be a complicated and difficult piece of work. The worm gear is shown, therefore, by drawing the pitch line of the teeth and other circles at the top and bottom of the teeth. At the point where the gear is in mesh with the worm, a portion

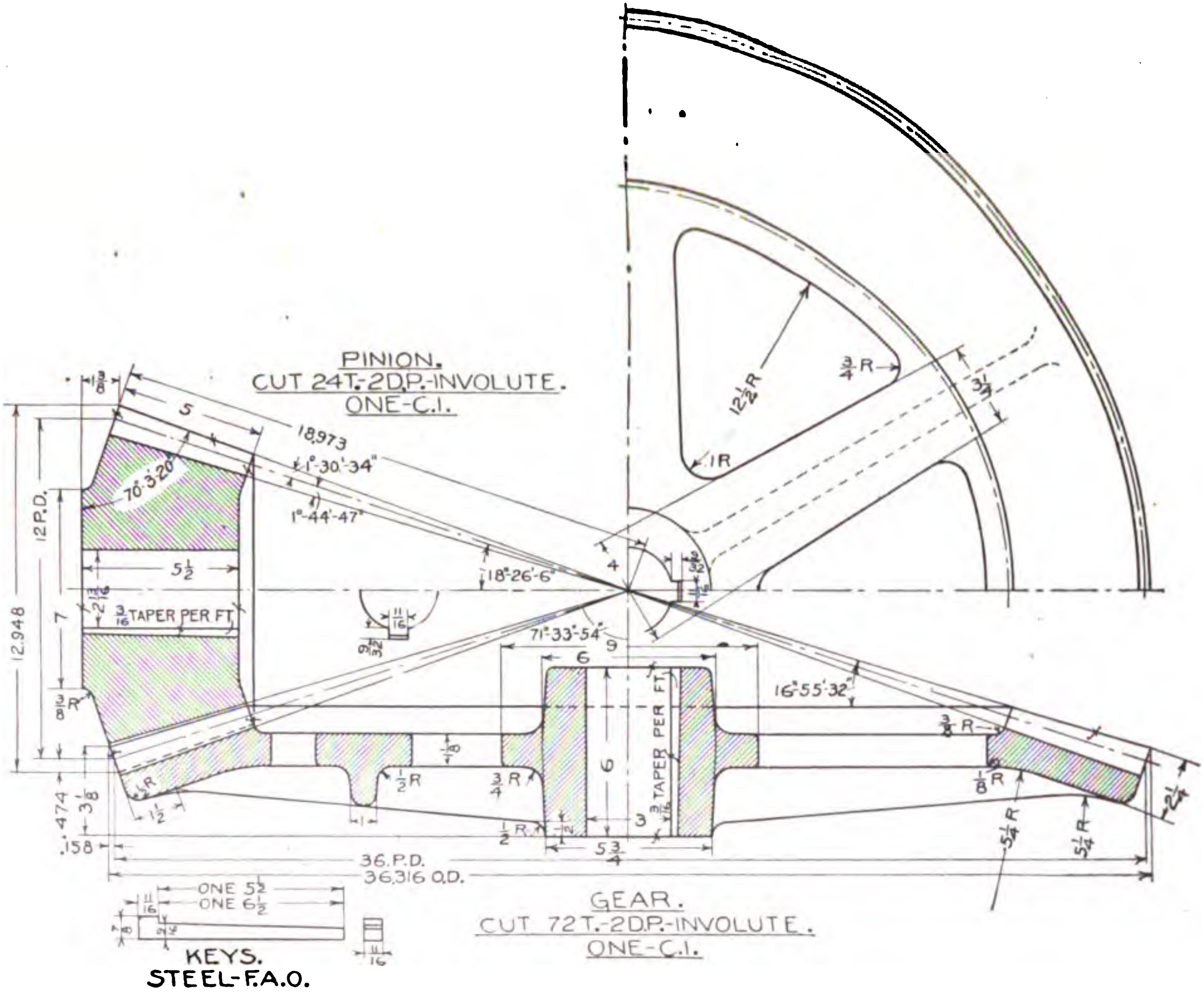


Fig. 42. Detail of Bevel Gear and Pinion.

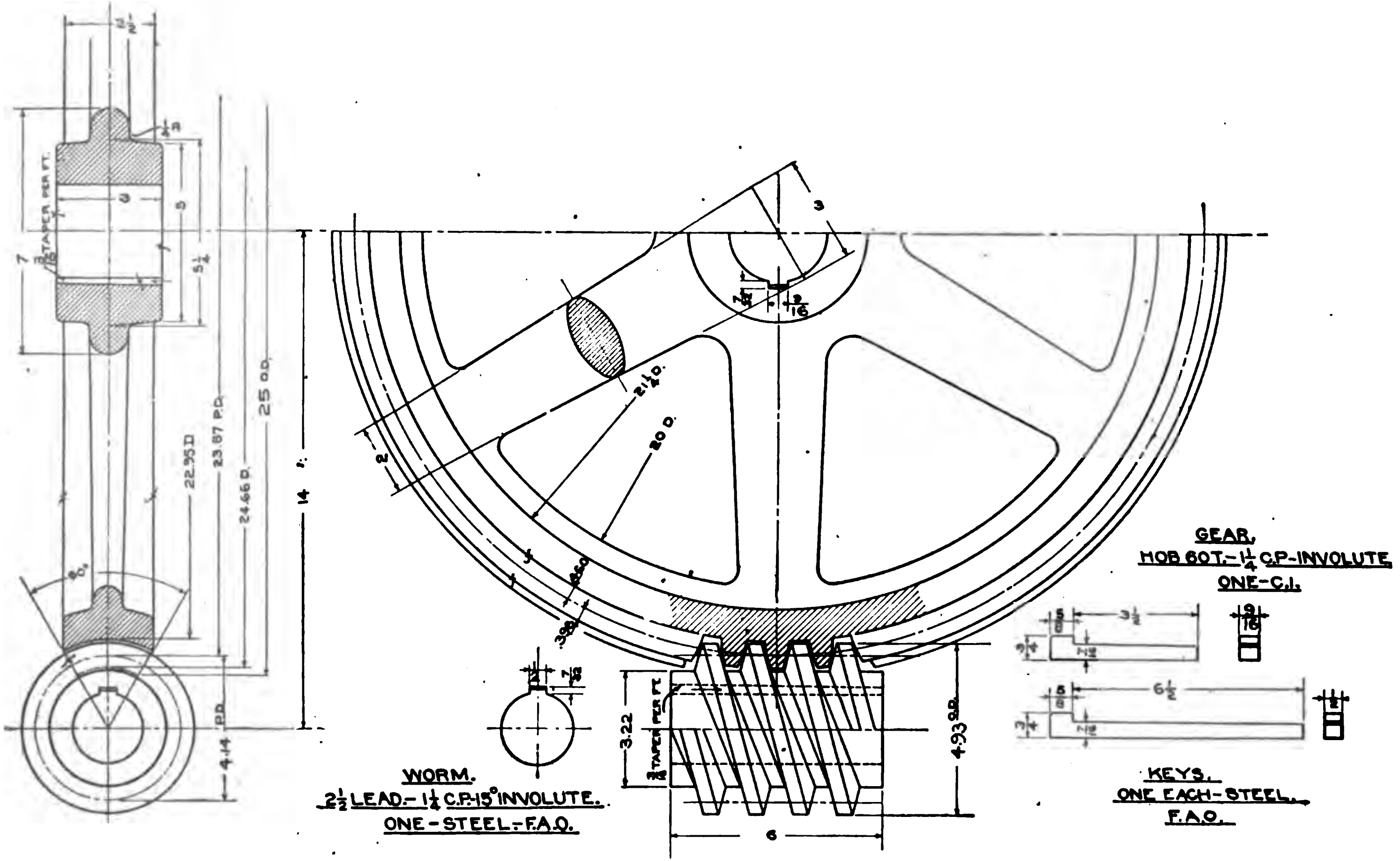


Fig. 48. Detail of Worm and Worm Gear.

of the rim is broken away, and the middle section shown. The worm threads, which are lines in the form of helices, are shown in the drawing merely by straight lines, this being the conventional way for representations of this character. Another way of showing the worm would be in cross-section, in which case the helical lines would be wholly avoided; as drawn, however, it has the advantage of at once conveying to the eye that it is a right-handed thread.

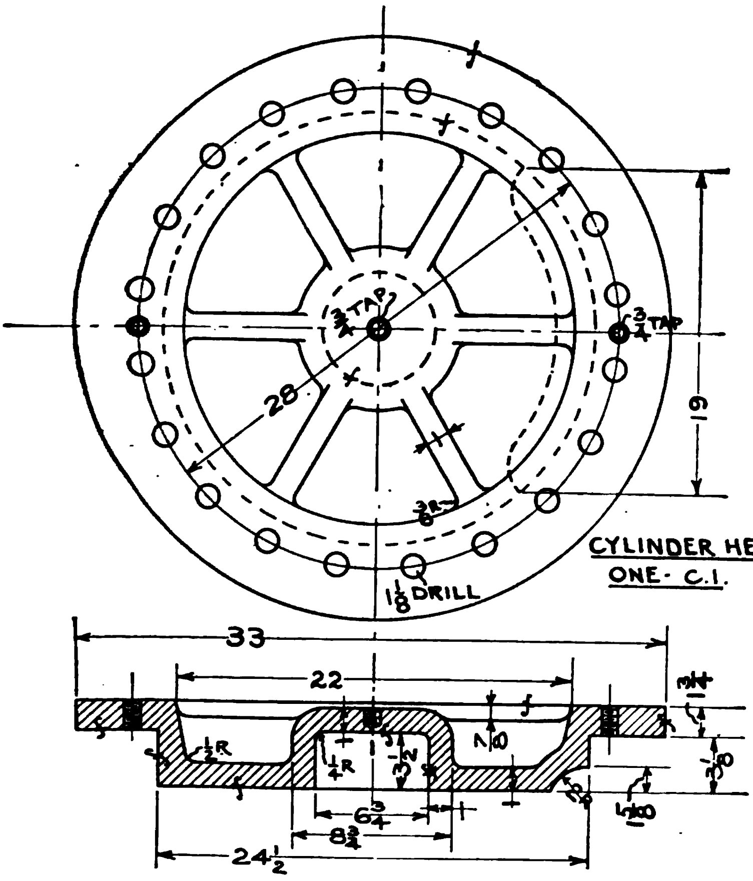


Fig. 44. Detail of Cylinder-head of Steam Engine.

Fig. 44 shows the cylinder-head of a steam engine. Although in the drawing the entire circle is shown, it would have been just as clear if only one-half had been shown, similar to the manner of showing the side views of the gears just discussed. In the plan view, it should be noted that the tapped holes are indicated by double circles, while the drilled holes show a single circle. The inner circle for the tapped holes is intended to represent the bottom of the thread, while the outer circle represents the top of the thread. Another conventional method for a tapped hole is to fill in the circle entirely with black ink; the method illustrated, however, is the most common.

Fig. 45 shows a water cylinder for a triplex pump, and is an excellent illustration of many of the points heretofore brought out, combined on a single drawing. This drawing should be carefully studied in detail. Note the general boldness of the lines, and the sharp contrast between the full-line work and the center, dimension, and dotted lines. Note that in the cross-sectional views the dotted lines have been almost entirely eliminated, thereby leaving the section work clear and plain. There is no necessity of showing parts in dotted lines which are beyond the plane of the section, but in several places just enough simple dotted work is shown to convey the relation of the parts.

Each sectional view is made for a specific purpose, and that purpose is never allowed to be obscured. The top plan is mainly to show the upper portion of the water cylinder; the cross-section below it gives the principal interior view; the right-hand half-elevation and cross-section is for the purpose of showing the face of the valve chamber and the interior of the valve chamber; the horizontal cross-section in the upper right-hand corner of the drawing, shows not only the interior of the barrel and valve chamber at two different points, but also a clear outline of the base of the entire casting. Each view must be used with the other to get a clear idea of the construction; but each view is so simple in itself that no confusion arises in the mind as to what its lines mean; one view is readily associated with the other, and the grouping of the four views is such that the eye passes easily over all of them.

Note the grouping of the dimensions, following in general the purposes of each of the views as explained above. On the top plan are given the dimensions affecting the top of the casting only. On the principal vertical cross-section are given the greater part of the dimensions for the entire piece. This is as it should be, for

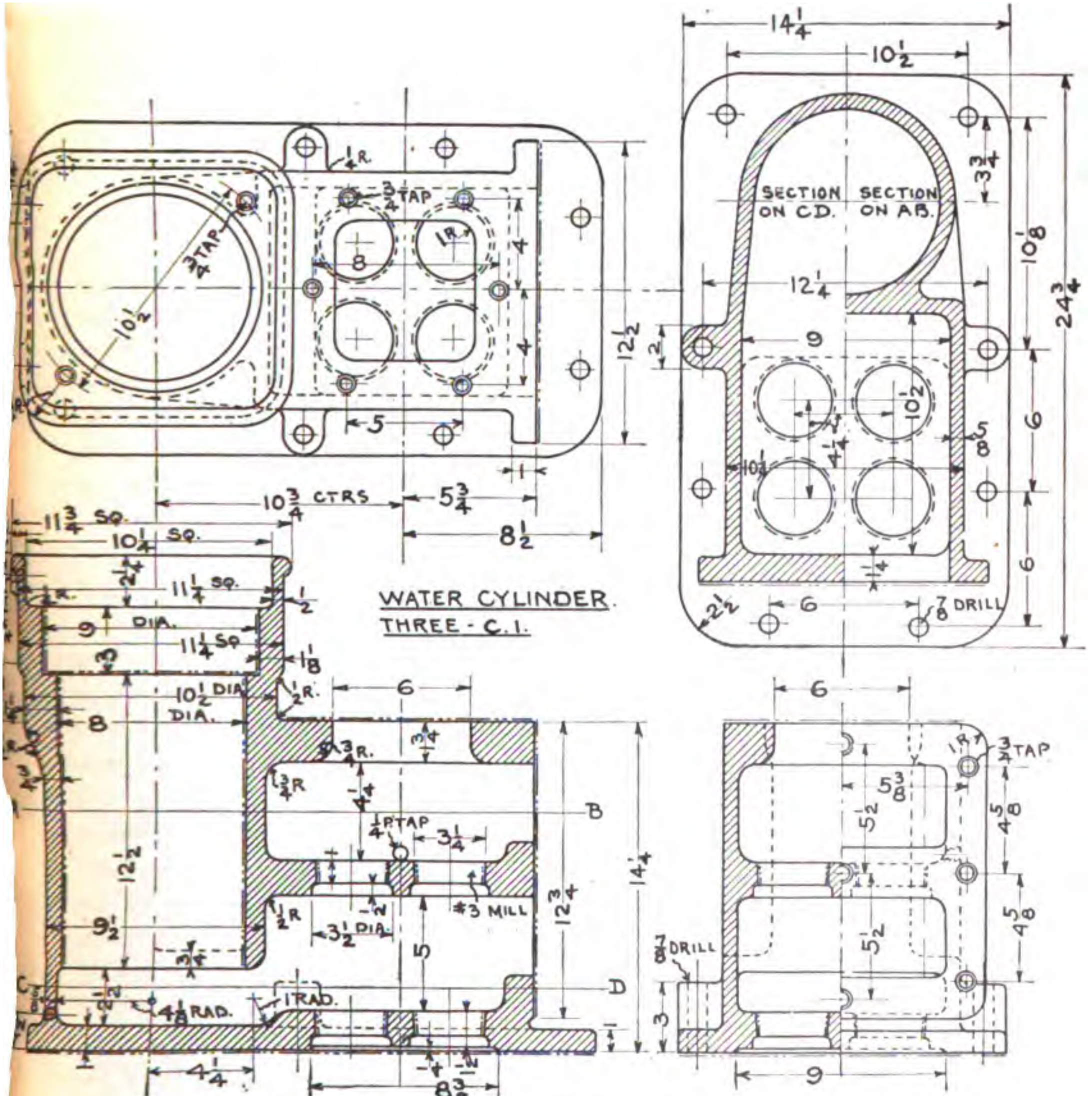


Fig. 45. Detail of Water Cylinder for Triplex Pump.

the dimensions should always be grouped as much as possible on the principal view of an object, provided they can be clearly put on that view and not become so numerous as to cause confusion. A drawing over which the eye has to wander widely in search of the several dimensions of the same portion, is slow and difficult to read. On the right-hand half-elevation and cross-section are the figures for the outline of the face of the valve chamber, and the location of the tapped holes for the hand-hole cover-bolts. On the horizontal cross-section are given the figures for the interior dimensions of the valve chamber, and a complete dimensioning of the base of the casting. The special attention of the student is called to this systematic grouping of the dimensions on the view which will most clearly show them. A glance at this drawing is sufficient to suggest what a confusion of figures there would be, had it been attempted to place them all on two views, and if, instead of cross-sections full and dotted lines had been used.

A new method is indicated on this drawing, of showing finished surfaces. Each surface which is intended to be finished in the machine shop, has drawn next to it a medium-weight line consisting of a long dash and two dots. This method of showing finished surfaces is not as common as the one heretofore used, of writing the letter "f" across the line; it has the advantage, however, of conveying an absolutely definite idea of the extent of the surface to be finished, and in some instances is especially valuable on this account. It is a good way of specifying the finish; but for general practice the letter "f" is simpler and perhaps more readily and universally understood.

This drawing, while not complicated, contains quite a large number of dimensions, and is a good example of the principle of systematic figuring. The student's attention, therefore, is called to the following analysis of the dimensions on the drawing.

The casting consists of a barrel, in which the plunger slides, with a stuffing-box at the top and a water-way at the bottom leading into the valve chamber; attached to this barrel is the valve chamber, consisting of two compartments, the lower one for suction, the upper one for discharge; to support both barrel and valve chamber and permit of their being bolted to the water-supply casting, a rectangular base is provided.

Beginning at the top of the casting, the figures for the stuffing-box, inside and outside dimensions and thicknesses are given, and note made that the outline is square. The tapped holes for the gland studs, and bosses for the drips, are shown most clearly in the top plan, and are therefore dimensioned there. Next we come to the bore of the barrel to receive the plunger, and here the square shape of the casting changes to a round, the diameter and thickness of metal being given. Below this cylindrical part, is the water-way, the height of which (2½ inches) is given, and then the 1-inch thickness of the base below. This completes practically all the dimensions of the barrel and stuffing-box.

Passing to the valve chamber, it is first necessary to locate the center line of same in reference to the barrel (10¾ inches, centers). This being done, the arrangement of valves is dimensioned, and figures given for the valve chambers, thus—length, breadth, depth, thickness of metal, fillets, etc.; then follow the location of the face of the valve chamber, 5¾ inches from the center line, and the layout for the hand-hole cover; then the location of the upper face of the valve chamber, 14¼ inches from the base, and the layout for the flange of the discharge pipe, which is shown on the top plan.

Few figures as yet have been placed on the base of the water cylinder; these are now completed by starting at one side of the base and going completely around same, giving not only external dimensions and radii, but also location of bolt-holes and their sizes—all of which are shown in the horizontal section.

This completes the dimensions; and if the student has carefully adhered to each particular part of the casting until completely dimensioned, and has not passed in haphazard fashion from one portion of the casting to another, he will have succeeded in dimensioning the piece with absolute completeness. No part will have escaped being dimensioned, and no part will be dimensioned twice. It would be a good plan for the student to copy this drawing, using a scale of 3 inches to the foot, and, in making the drawing, to follow the description as given above in reading the figures from the cut. He will thus more clearly realize the systematic progress from one part of the casting to the other, and will himself check the figures shown.

Fig. 46 shows the detail of a hoisting drum to carry wire rope. Attention is called in this detail to the enlarged cross-section of the rim, conveniently placed to show clearly the style of the groove. It should also be noted, that, instead of drawing the grooves the entire length of the drum, but a few are drawn at each end of the drum, and a note placed against same to indicate that the grooves are to be cut the entire length. This is another "short cut" consistent with the definition of a working drawing. The breaking away of a portion of a view is illustrated in the right-hand elevation, in which a small section is exposed to show the method of fastening the end of the wire rope.

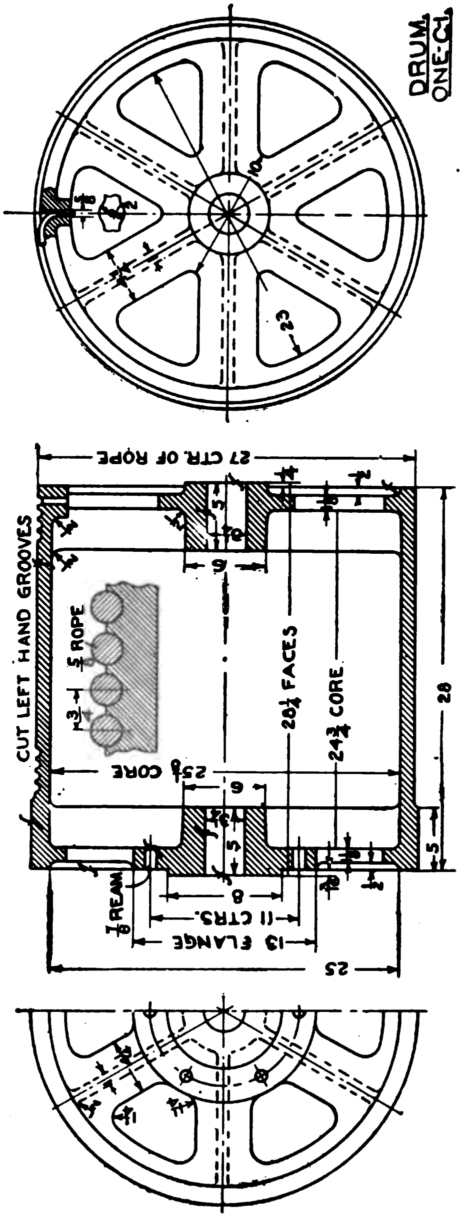


Fig. 46. Detail of Hoisting Drum to Carry Wire Rope.

Fig. 47 shows a crane drum, grooved for chain, and carrying its driving gear and shaft. This is a very good illustration of the economy of grouping parts together, instead of detailing them separately. It is obvious that the drawing of each detail is just as clear as though it were separately drawn. In fact, the information conveyed is the most complete possible; for not only are the figures for each part clearly shown, but the exact manner in which the

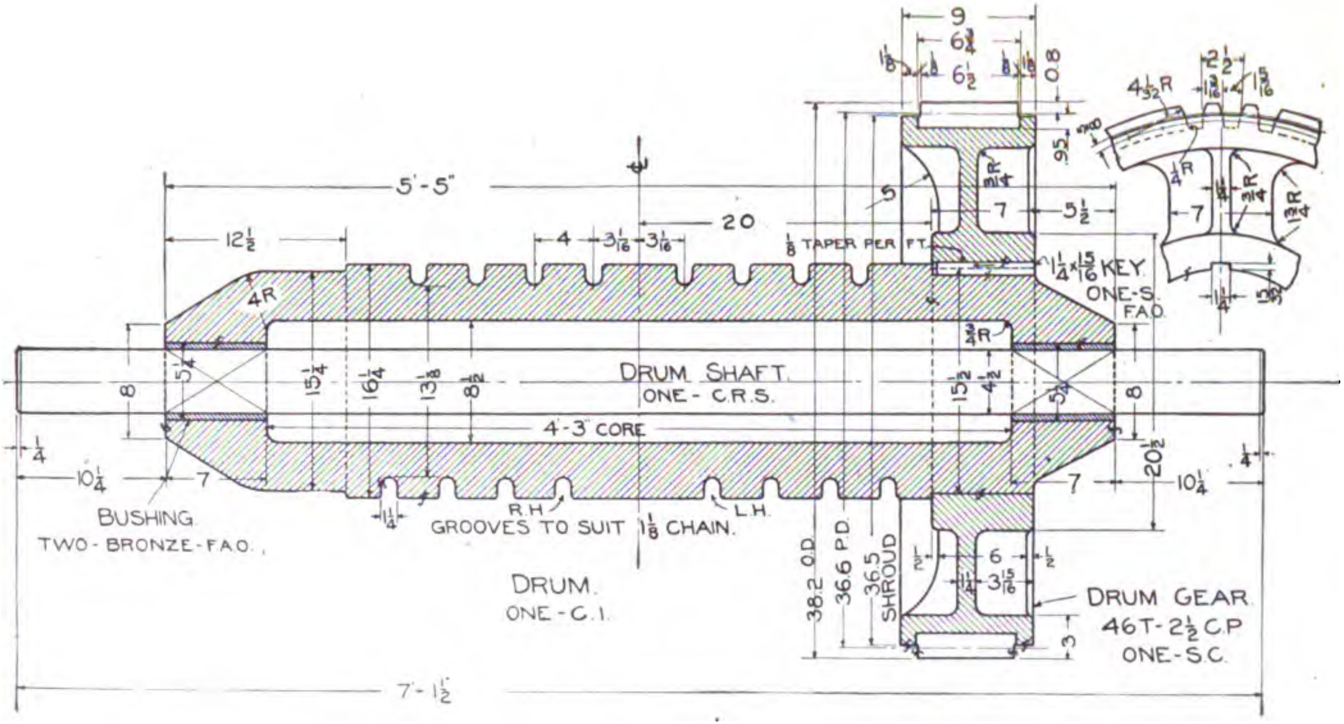


Fig. 47. Detail of Crane Drum.

parts go together, thus enabling the workman to understand at a glance the assembling of the parts, and to make his fits accordingly. A general drawing for this purpose alone has to be made in the case of a complicated machine; and it would be confusing to put on such a drawing figures sufficient to detail each part. The draftsman who can properly judge when to use assembled drawings for detail dimensions, and when to avoid such use, will save a large amount of time and money in the production of drawings for shop use. A common rule that "every part shall be detailed separately" is in vogue in many drafting rooms; but it is seldom followed literally, and when so followed becomes a drag on office efficiency. A better rule is—"Detail every part separately when groups of parts cannot be clearly detailed together."

Note on this drawing the method of indicating, by light diagonal lines across the shaft, the location of the bearings; also the enlarged view of a few teeth of the gears, with sufficient figures for the pattern maker to work out the teeth. These gears are "half-shrouded," or strengthened by a rim extending up to the pitch line.

Fig. 48 shows the detail of an ordinary shaft with a number of different bearings and fits upon it. It illustrates most of the common points which are necessary to be specified on shaft details. The

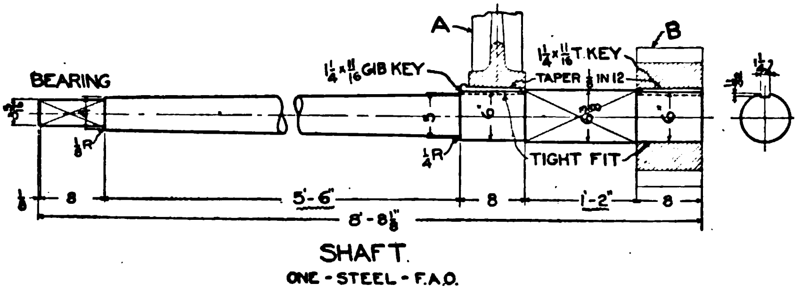
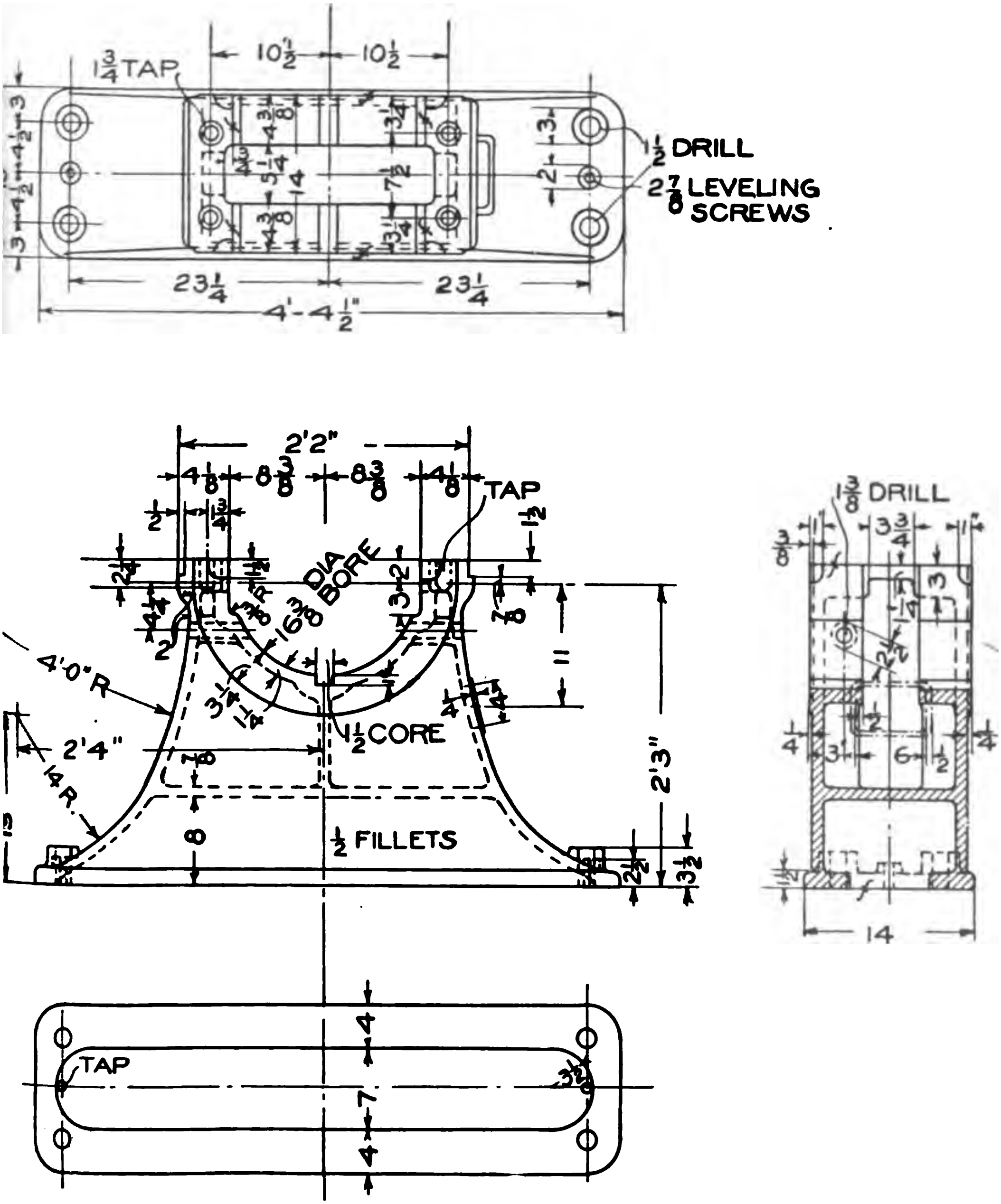


Fig. 48. Detail of an Ordinary Shaft.

distance, shoulder to shoulder, is usually given throughout the entire length of the shaft, and these distances summed up for the "over-all" dimension. The "over-all" dimension is important, because from it the stock is ordered and cut off; and the workman should not be required to add up a lot of figures to secure it. Fillets should be allowed at every shoulder, if possible, and their radius specified;



BEARING STAND.

ONE-C.I.

Fig. 49. Detail of Bearing Stand with Cap and Boxes Removed.

in this way there is less liability to the formation of incipient cracks than if the corners are left sharp. Key-ways should be carefully dimensioned and located. Bearings should be indicated by light lines running across the shaft diagonally; and it is good practice to print the name of each piece to which the shaft is fitted, just above the shaft at the point where such fit occurs, or the parts themselves are partially shown in light lines, as at A and B in the figures, thus enabling the workman to make the fits more intelligently. It is common practice to make all holes which receive shafts of exactly "gauge diameter," and to make the allowance for the fit in the shaft. For example, a "3-inch running fit" would mean that the hole in the piece to receive the shaft would be exactly 3 inches in diameter, while the shaft would be, say, "3 inches less .003 inch." Sometimes this allowance is indicated by giving the actual number of thousandths of an inch under size, as noted; sometimes by calling for a "running fit," or a "wringing fit," or a "pressed fit," or a "drive fit," or a "tight fit," as desired.

Shafts are often so long that it is difficult to represent their entire length on the sheet to the scale chosen. They are then "broken," as shown in the figure, and crowded up to a shorter length, the dimensions being depended upon to give the proper relation of the parts.

When there is occasion, because of some change, to alter a dimension on a finished drawing, it is usually permissible to change the dimension without rubbing out the lines of the drawing, provided that no considerable number of other dimensions are affected, and provided that some sign or note is made on the drawing, calling attention to the fact that the dimension has been changed and that the drawing is "out of scale." Sometimes the dimension is placed in a circle thus ⑥" or a line drawn beneath it thus, $\underline{6''}$; or the words "out of scale" placed after it thus, 6" (out of scale). Although workmen are not allowed to "scale" drawings, yet it is dangerous to have dimensions which are out of scale on the drawings unless special attention is called to that fact.

The above remarks on "broken" pieces and "out-of-scale" dimensions are equally applicable to all details as to shafts, the

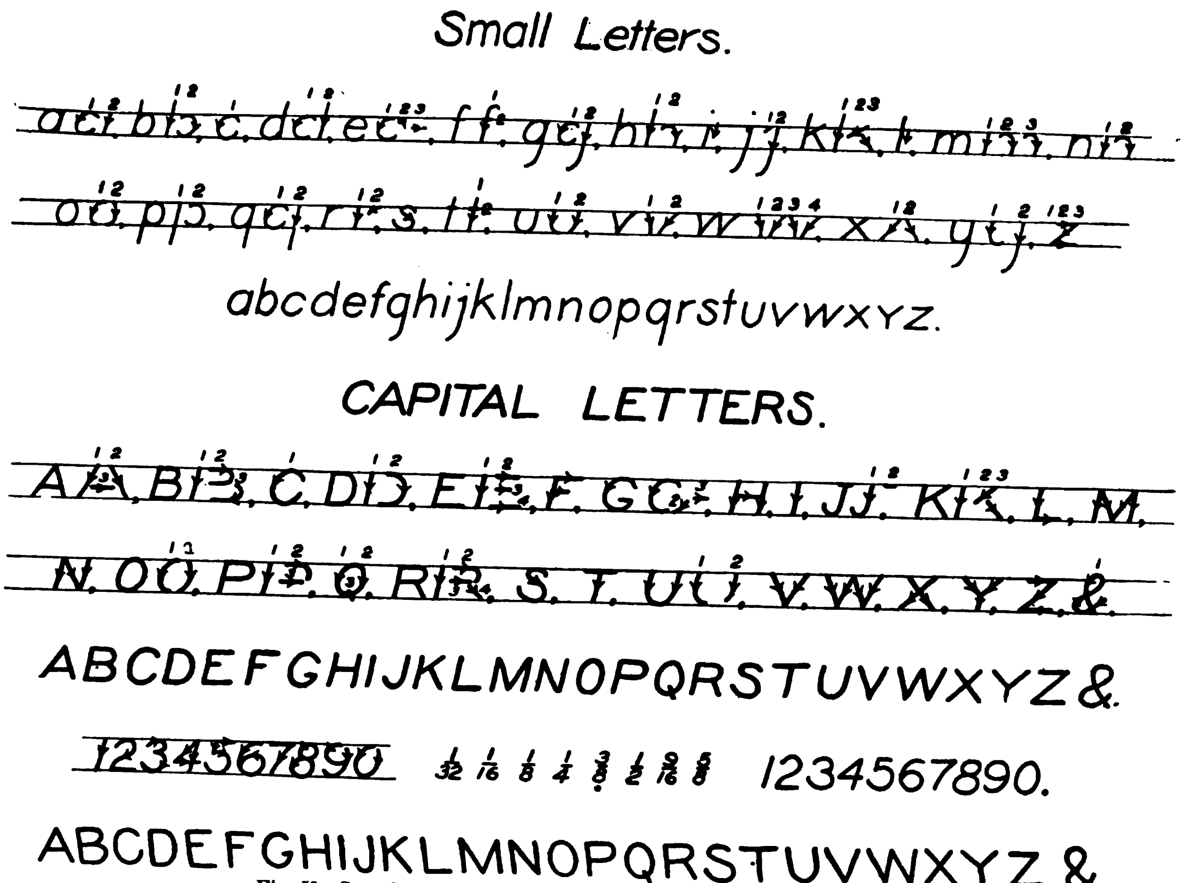


Fig. 50. Sample Sheet of Plain Letters Used in Working Drawings.

points merely being illustrated by the figure under discussion.

Fig. 49 shows a bearing stand with the cap and boxes removed. There is little of special note to discuss in regard to this, beyond calling attention to the general nature and type of the piece illustrated. The design is characteristic of pedestals and bearings found about stationary engines of large size. Such parts are usually massive and heavy in their proportions, with well-rounded corners and smooth outline. The closed-box form of casting affords maximum strength with good distribution of material, and at the same time conveys to the eye the effect of a solid piece throughout.

Fig. 50 shows a sample sheet of plain letters, such as are particularly applicable to working drawings. They are especially devised for easy, quick, and uniform strokes. Each draftsman has a character of his own in lettering and figuring, and the form of lettering which is most natural for him to use is the one he will use to best advantage. It is necessary, however, to confine draftsmen to a general type in order to make their work reasonably uniform; and the sample sheet (Fig. 50) represents not only the most common type in use, but a type to which almost any draftsman can readily train his eye and hand. Whether the slopes are forward or backward, or straight up and down is of little importance, as long as the general style is maintained.

Every drafting room has certain methods and rules peculiar to its own organization and that of the shop to which it supplies drawings. While it is impossible to formulate any set of instructions which will cover all situations, the accompanying sheet, "Drawing Office Practice," is consistent with general practice in modern drawing offices and the fundamental principles discussed elsewhere in this book. It is a condensed code of procedure which the student will do well to hang in some convenient location near his table, and to consult freely as he works. It should not take the place of the explicit discussions of the text, but should be used as an index to it and as a reminder.

DRAWING ROOM PRACTICE

CHICAGO AMERICAN SCHOOL OF CORRESPONDENCE ILLINOIS

STD. SIZES

	TRIMMING SIZE	TITLE FRAME
PLATES	9 × 12	1 × 2
	12 × 18	1 × 2
DRAWINGS	18 × 24	$2\frac{1}{2}\times 4$
	24 × 36	$2\frac{1}{2}\times 4$

MARGINS. ALL MARGINS $\frac{1}{2}$" WIDE

SCALES. USE THESE SCALES IN PREFERENCE TO OTHERS

12" = 1' FULL SIZE

6" = 1' HALF "

3" = 1' QUARTER "

$1\frac{1}{2}$" = 1' EIGHTH "

CYLINDER
FOR
12×36 CORLISS ENGINE
AMERICAN SCHOOL OF CORRESPONDENCE
CHICAGO ILLINOIS
SCALE 3" = 1' SEPT. 1. 1906
C. L. G.

(SAMPLE TITLE)

CHOOSE LARGEST SCALE CONSISTENT WITH SIZE OF SHEET.

PROJECTION USE SIMPLE PROJECTIONS IN THIRD ANGLE ONLY.

MAKE VIEWS TO COMPLETELY ILLUSTRATE · NO MORE · NO LESS!

PLACE VIEWS ON SHEET IN SAME POSITION AS PIECE OCCUPIES IN ASSEMBLED MACHINE.

WORK ALL VIEWS TOGETHER. DO NOT TRY TO FINISH ONE VIEW BEFORE BEGINNING ANOTHER.

DOTTED LINES. USE FEW DOTTED LINES. ONLY WHEN ABSOLUTELY NECESSARY.

USE CROSS SECTIONS FREELY, IN PREFERENCE TO DOTTED LINES.

PENCIL WORK. MAKE PENCIL DRAWING SHARP AND DEFINITE, ABSOLUTELY COMPLETE, AND CHECK CAREFULLY BEFORE TRACING.

TR. CLOTH. USE ROUGH SIDE OF TRACING CLOTH TO PREVENT CURLING.

CHARACTER OF LINES.

LINES FOR SIMPLE LARGE SCALE DETAILS THUS ━━━━━━━━

" " COMPLICATED SMALL " " ────────

" DOTTED " - - - - - - - - - -

" CENTER OR AXIAL { SOLID / DASH AND DOT " ──────── / ── · ── · ──

" DIMENSION " |←— $4\frac{1}{8}$ —→|

" CROSS SECTION $\frac{1}{16}$ TO $\frac{1}{8}$ APART ▨

COLOR. USE BLACK INK FOR ALL LINES

FIGURES. MAKE FIGURES BROAD, BOLD AND ABSOLUTELY CLEAR.

MAKE ALL FIGS. READ FROM LOWER OR RIGHT HAND SIDE OF SHEET.

DIMENSIONS MAKE ALL DIMENSIONS IN INCHES UP TO AND INCLUDING 36 IN.

INCH MARKS (") ARE SELDOM NECESSARY, EXCEPT CASES AS 3'-6".

ANY DIMENSION NEED OCCUR BUT ONCE ON SAME DRAWING.

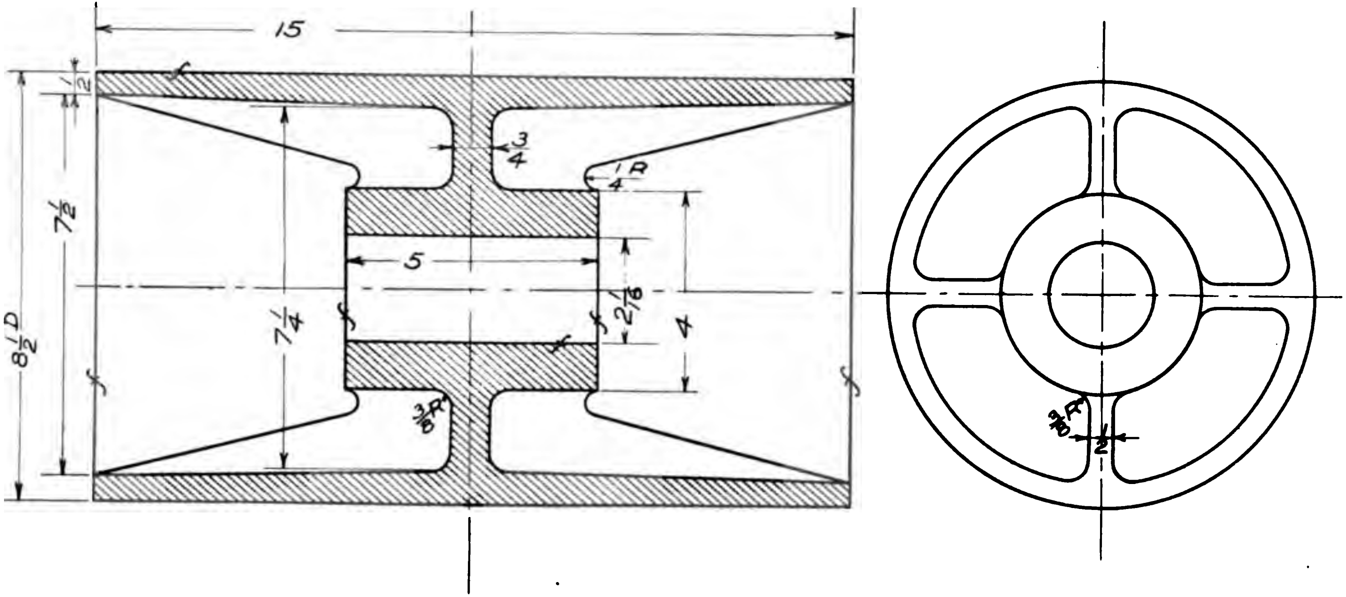
USE ENOUGH DIMENSIONS TO ENABLE THE PIECE TO BE MADE · NO MORE. NO LESS!

NOTES. USE NOTES FREELY TO CLEAR UP DOUBTFUL POINTS.

THOUGHT. ALWAYS HAVE IN MIND THE WORKMEN WHO ARE TO USE THE DRAWING · THE PATTERN MAKER, BLACKSMITH AND MACHINIST.

ALWAYS CONSIDER THE MACHINES AVAILABLE FOR THE WORK.

PLATE XVII



PLUNGER
TWO C.I.

APRIL 14 1907 HERBERT CHANDLER CHICAGO. ILL.

PLATES.

Plates XVI to XIX inclusive are to be made by the student as an examination on the work of Part IV. These plates are to be the same size as those of the preceding Instruction Papers of the course, viz., 11 inches by 15 inches outside, with a margin of $\frac{1}{2}$ inch, making the clear space for the drawing 10 inches by 14 inches.

PLATE XVI.

Sketches. From plates XVII and XVIII make rough, freehand sketches of the parts, paying no attention to scale or arrangement on the sheet. Simply make them of such size as to get them all on the standard sheet as specified. Use a medium pencil, and try to make the sketches quickly. Put on all the dimensions. Do not try to get smooth, straight lines, but dash off boldly with the pencil, aiming merely to get clearness and definiteness. Make the sketches over two or three times, noting any improvement in rapidity or effectiveness, and submit the best one.

PLATE XVII.

Water Plunger. This represents the plunger for a water cylinder of an ordinary pump. Make the drawing to a scale of 6 inches to the foot, and instead of copying the plate, use the sketch made on Plate XVI. This will not only give practice in working from a rough sketch, but will show whether all the information is on the sketch.

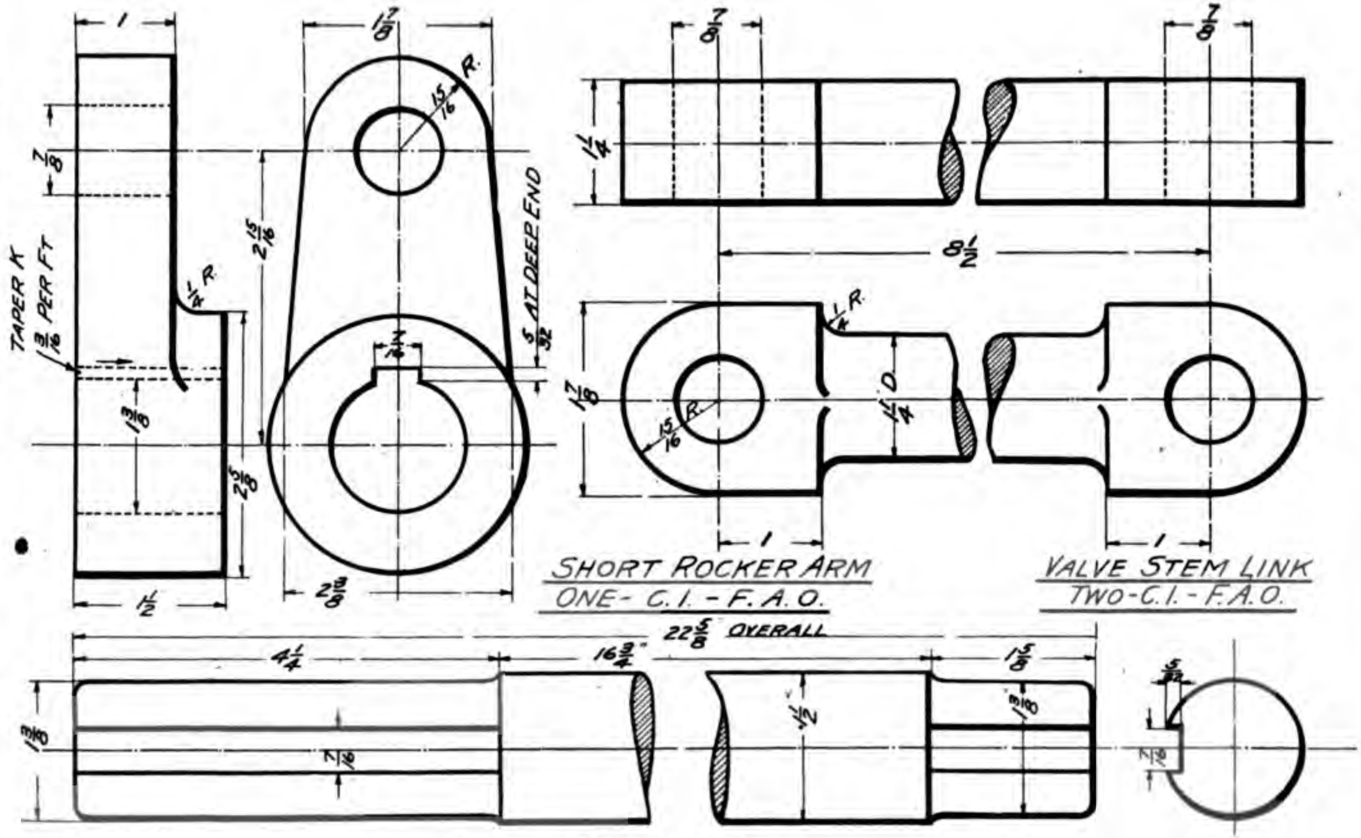
PLATE XVIII.

Valve Motion Details. This represents some of the details of the valve motion of an ordinary pump. Make the drawing to a scale of 12 inches to the foot, or full size, and as before, instead of copying the plate, use the sketches made on Plate XVI. Note that this plate requires the shade lines to be put on. This being a full size drawing, will stand very bold heavy lines, thus insuring a good blue print. This plate is to be made on tracing cloth, laid over the pencil work.

PLATE XIX.

Water Valve Details. This represents the complete valve mechanism for the water cylinder of an ordinary pump. Make the drawing to a scale of 12 inches to the foot, or full size. Although it is evident from the plate that this is a case where the dimensions can be placed on the assembled drawing without confusion, yet in order to give practice in detailing from an assembled drawing, the student is required to detail each part separately in the space to the right of the assembled valve as shown. The assembled view should first be made, copying the plate. Then, instead of placing the dimensions on the assembled view as shown, they should be placed on each individual detail as drawn. There are five parts to be shown, and the arrangement of these on the sheet, and the number of views necessary for each is left entirely to the student. This is a good opportunity to produce a nice, well filled sheet, workmanlike in execution and arrangement, by exercising careful attention to the principles of the foregoing pages.

This plate should be made on tracing cloth laid over the pencil work.



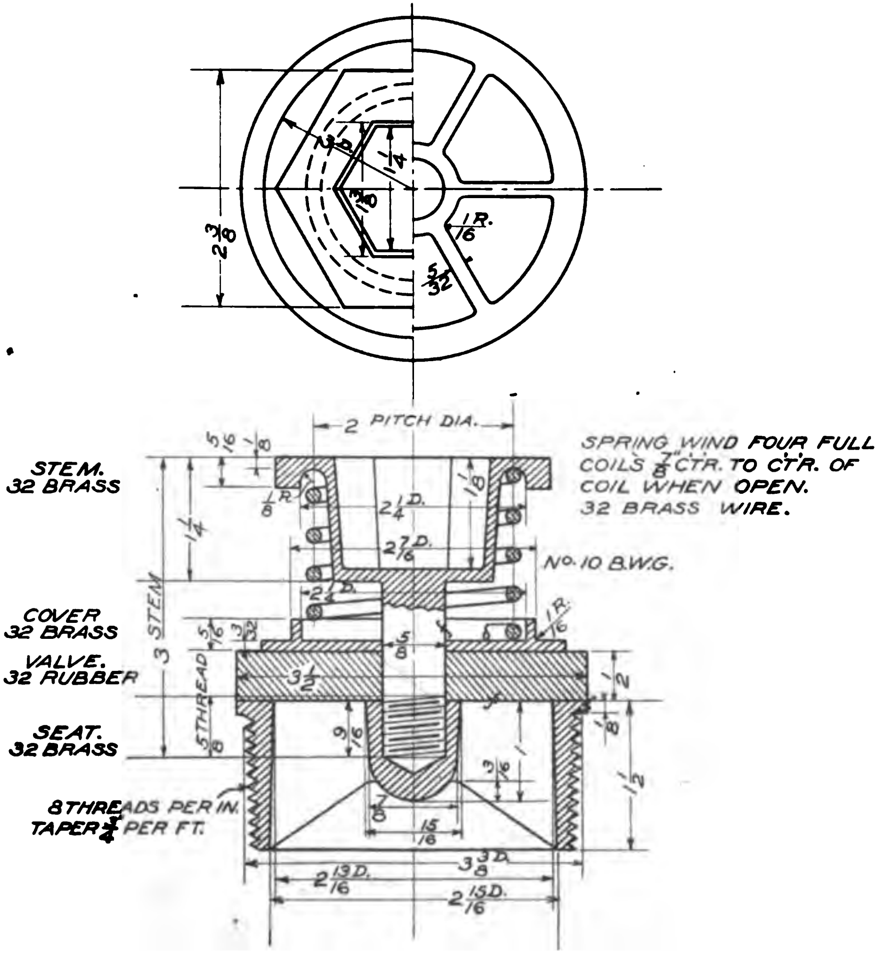
SHORT ROCKER ARM
ONE- C.I. - F.A.O.

VALVE STEM LINK
TWO-C.I. - F.A.O.

UPPER ROCKER SHAFT
ONE-STEEL-F.A.O.

APRIL 27, 1907 HERBERT CHANDLER CHICAGO, ILL.

PLATE XIX


2 PITCH DIA.
SPRING WIND FOUR FULL COILS 7/8" CTR. TO CTR. OF COIL WHEN OPEN.
32 BRASS WIRE.
No. 10 B.W.G.
STEM.
32 BRASS
COVER
32 BRASS
VALVE.
32 RUBBER
SEAT.
32 BRASS
8 THREADS PER IN.
TAPER 3/4 PER FT.
3 STEM
5 THREAD
2 3/8
1 1/4
1 R/16
5/32
1 3/8
1/4
1/10
5/16
1/8
1/8 R.
1 1/4
2 1/4 D.
2 7/16 D.
1 1/8
2 1/4 D.
1 R/16
5/16
3/32
5/8
3 1/2
1/2
1/8
1 1/2
9/16
1
3/16
7/8
15/16
3 3/8 D.
2 13/16 D.
2 15/16 D.

MECHANICAL DRAWING

PART V

MECHANISM DRAWING

In Part IV Working Shop Drawings have been analyzed in detail, systematic processes for making them have been outlined, and numerous illustrations given and thoroughly discussed. The student as yet, however, has not been shown how to originate the theoretical outlines of the surfaces controlling motion in machines; and it is the purpose of Part V to accomplish this.

The theoretical shape of the working surfaces of a machine can be studied and developed to best advantage without any consideration as to their strength or their ability to perform work or withstand service. Such study is a study of the *mechanism* of a machine, and must always precede the study of design to provide the proper strength.

A *mechanism*, therefore, is a combination of parts so formed and connected as to produce a desired motion, but not necessarily to perform any specific work.

A *machine* is a working mechanism, or a combination of mechanisms, suitably designed for the performance of specific work.

Mechanism drawing is really the first step in Machine Design; and all the familiar parts of machines, such as springs, screws, cams, pulleys and belts, gears, etc., are dependent for their existing practical form upon their theoretical layouts as mechanisms, involving exact mathematical principles. The student should pursue carefully the study of motion as applied to the development of the common machine parts, as it is fundamental to the advanced work which follows it.

THE HELIX

Since most coil springs and all screw threads depend upon a curve known as a helix, it will be necessary to know what a helix is, and how it can be drawn, before taking up the construction of the springs and screws.

Suppose we take a cylindrical piece of wood, such as is shown in Fig. 1, and a rectangular piece of paper A B F E, with the side A B equal to the circumference of the cylinder, and the side A E equal to the length of cylinder. If we lay off along A E any convenient distance A D, and draw the line D C parallel to A B, we have the rectangle A B C D. Now draw the diagonal A C of this rectangle and wrap the paper around the cylinder, keeping the side A E on an element of the cylinder; the paper will just cover the cylinder, the edge B F meeting the edge A E. The point C coincides with the point D, and is on the same element of the cylinder as A; therefore the line A C has made one complete turn around the cylinder, advancing the distance A D in this turn. The curve which the line A C now takes is called a *helix*, and the distance A D is called the *pitch of the helix*.

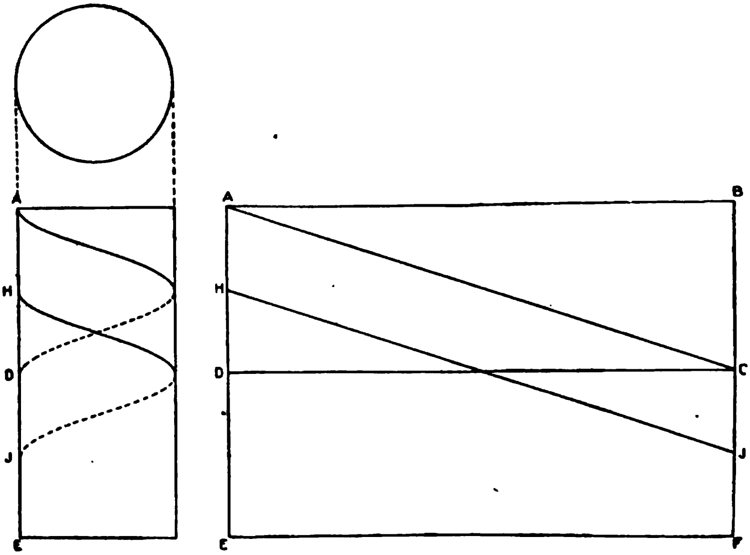


Fig. 1. Helical Curve and Development.

If on the piece of paper we also choose a point H, half way between A and D, and draw from this point a line H J parallel to line A C, this line H J will form another helix parallel to the helix formed by the line A C, when the paper is wrapped around the cylinder. The pitch of both helices is the same.

The helix is often incorrectly called a spiral, but there is a marked difference between the two. The *spiral* is a curve which lies in one plane, and winds around a point, constantly receding from the point, according to some law. The mainspring and hairspring of a watch are in the form of spirals.

To draw the projections of a helix we must know the diameter of the cylinder upon which the helix is formed, and the pitch of the helix. Fig. 2 shows the construction. Draw two projections of the cylinder A B D C; along any element, preferably one of the contour elements A B or C D, lay off the pitch A E. Divide the circumference of the circle, which is the end view of the cylinder, into any number of equal parts, and number the points of division 1, 2, 3, etc. Divide the pitch A E into the same number of equal parts, and number these points of division in the same way that the points on the circle are numbered, calling the point A No. 1. From point 2 on the circle, draw a line parallel to A B; and from point 2 on A B, draw a perpendicular to A B. The point L, where the parallel line meets the perpendicular line, is one point on the projection of the helix. The points M, N, etc., are found in the same manner. A smooth curve starting from A, going through all the points and ending at E, will be the projection of one turn of the helix. The half from A to R is on the front, and is therefore a full line, while the half from R to E is on the back and is a dotted line. It should be observed that the point R is on the perpendicular from 7, which is just half way between A and E; that is, the distance

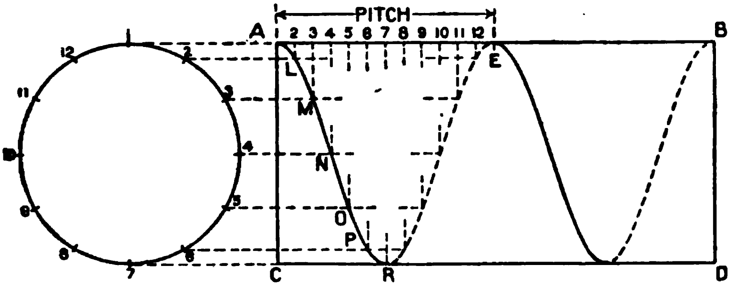


Fig. 2. Right-Hand Helix.

C R is just one-half the pitch. The curve from E to B is the projection of the next turn of the helix and is exactly like the first one.

The helix shown in Fig. 2 is called a *right-hand* helix. If the curve starts at C and is drawn as in Fig. 3, we have a *left-hand* helix. Notice that the visible part (from C to R) slants in the same direction as the invisible part of the right-hand helix, which is shown dotted in Fig. 2.

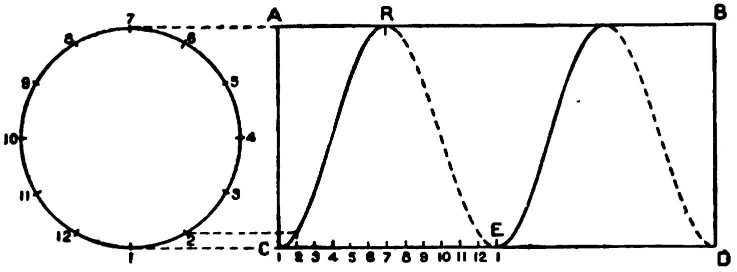


Fig. 3. Left-Hand Helix.

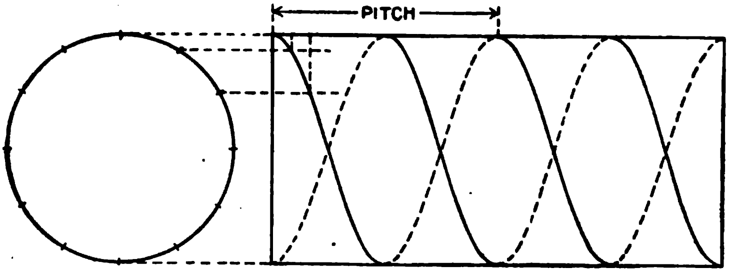


Fig. 4. Right-Hand Double Helix.

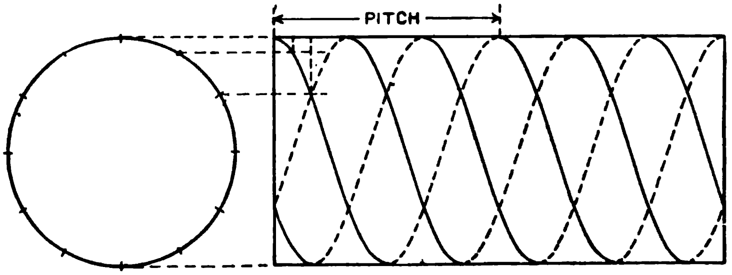


Fig. 5. Right-Hand Triple Helix.

Since the helix is a line drawn on the surface of a cylinder, the other projection of the helix must be the circumference of the circle, which is the end view of the cylinder. Fig. 4 shows a right-hand double helix, and Fig. 5 is a right-hand triple helix.

The construction of these curves should be studied carefully in order that the springs and screw threads may be better understood.

Helical Springs. If, instead of winding a line around a cylinder in the form of a helix, as shown in the preceding figures, we wind a piece of spring wire, we shall get a helical spring.

Fig. 6 is the drawing of a helical spring of round wire, the side view only being drawn, as this is all that is necessary to give all the dimensions. To draw the spring we must know the pitch, the diameter of the wire, and either one of the dimensions A, B, or C. If A is given, we subtract from A the diameter of the wire to find B; and if C is given, we add to C the diameter of the wire to get B; then, knowing B and the pitch, we can draw the helix (which is shown in dotted lines) exactly as the helix was drawn in Fig. 2. This is the helix formed by a line in the center of the wire. Now draw a series of circles with centers on this helix and of a diameter equal to the diameter of the wire. Smooth curves drawn tangent to these circles, as shown in the figure, will give the projection of the spring.

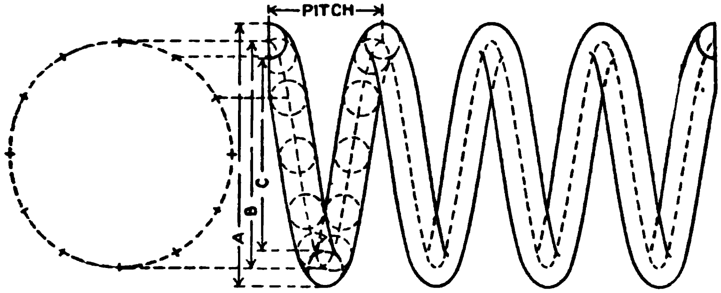


Fig. 6. Helical Spring, Round in Section.

Fig. 7 shows a helical spring of square wire. The drawing of this is simply the drawing of four helices, starting from each of the corners of the square A C M L; this square being the cross-section of the wire of which the spring is made. All four of the helices have the same pitch, equal to A B for, since the square B D P N is the same as A C M L, the distance C D is the same as A B; and since the points L, M, N, and P are vertically under A, C, B, and

D, respectively, the distance L N is equal to A B, and M P is equal to C D. The helix A F B has a diameter equal to that of the circle I E, and is drawn by dividing the circle I E and the pitch A B, as in Fig. 2; and the helix C G D, having the same diameter as A F B, is drawn by dividing circle I E and pitch C D. The helix L H N has a diameter equal to that of the circle K R, which is I E minus twice the thickness of the wire, and is drawn by dividing up the circle K R and the pitch L N; and the helix M J P, having the same diameter as L H N, is drawn by dividing circle K R and pitch M P. Since the two circles are drawn about the same center, the divisions on circle K R can be found by drawing radial lines from the points of division on circle I E. The vertical lines drawn from the divisions of the pitch A B can be used for the divisions of L N; and those drawn from divisions of C D can be used for M P.

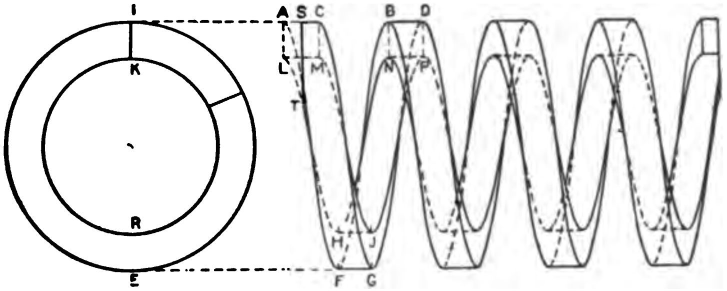


Fig. 7. Helical Spring, Square in Section.

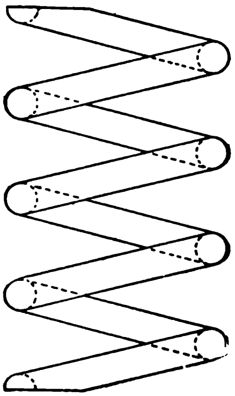


Fig. 8. Conventional Representation of Helical Spring, Circular in Section.

After the four parallel helices are drawn, it is necessary to study the drawing carefully, to decide what lines will be visible (full lines) and what invisible (dotted lines). Dotted lines should be used from H to J, N to P, etc., and full lines from F to G, B to D, etc. The line S T is the end of the spring, and consequently any part of a helix which goes outside of that line should not be left on the finished drawing. It is better, however, to draw in the whole of the square A C M L, and to draw the helices starting from A to L in order to draw those parts of the same helices which lie to the right of S T. The parts to the left of S T are shown in the figure by light, dotted lines to indicate that they are construction lines, and not a part of the projection of the spring itself.

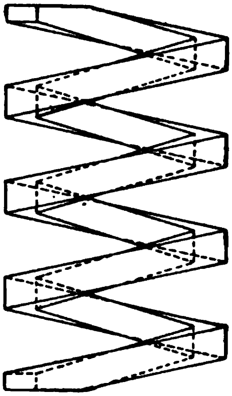


Fig. 9. Conventional Representation of Helical Spring, Square in Section.

Conventional Representations of Springs. To draw springs by the method just explained involves considerable work and would consume a great deal of time if many were to be drawn; therefore, in working drawings, the draftsman commonly uses a conventional method. This conventional drawing is similar to the true projection, except that straight lines are used in place of curved lines. Fig. 8 shows the conventional drawing of a spring of round wire; and Fig. 9, of a spring of square wire. Springs are often shown in half-section, as in Fig. 10, this method involving less work than the method of Figs. 8 and 9.

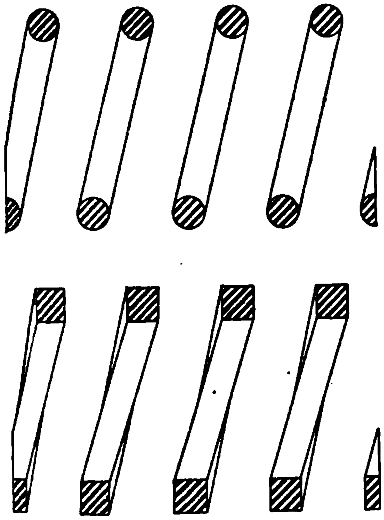


Fig. 10. Sections of Helical Springs.

SCREW THREADS

If we cut a groove around a cylinder in the form of a helix, we shall have what is called a *screw thread*, the thread being formed by the material which is left between the successive turns of the helical groove. A cylinder having such a helical groove cut around it is called a *screw*; and a piece having a cylindrical hole in it, with a helical groove cut around the hole, is called a *nut*. The most common uses of the screw are to fasten pieces together, hold them at a given distance apart, and to cause one piece to move with relation to another piece.

V-Thread. The form of screw thread with which we are most familiar is what is known as the V-thread, shown in its simplest form in Fig 11. Fig. 12 shows the method of drawing the true projections of this thread. The dimensions which must be known in order to make the drawing, are the outside diameter A O, the pitch A C, and the depth of the thread A K. First draw the two projections of a cylinder of a diameter equal to the outside diameter of the screw. Half of the end view is sufficient. On the line A E of this cylinder lay off A C equal to the pitch; starting at A, draw the helix A B C D, as described for Fig. 2. Inside of the cylinder A O, draw a smaller cylinder K L, the diameter of which is equal to the diameter A O minus twice the depth of the thread. Now, on this smaller cylinder, starting at point H, perpendicularly under a point on the line A C which is half way from A to C, draw the helix H J V with the same pitch as was used for the helix A B C. Draw the lines P R, X Y, S T, etc., tangent to the two helices and the projection of the thread is completed. It is necessary to draw the invisible parts of the two helices in order to draw the lines S T, X Y, etc.; but they need not be left on the finished drawing. In Fig. 12 they are shown dotted for one turn of the screw, in order to show the construction.

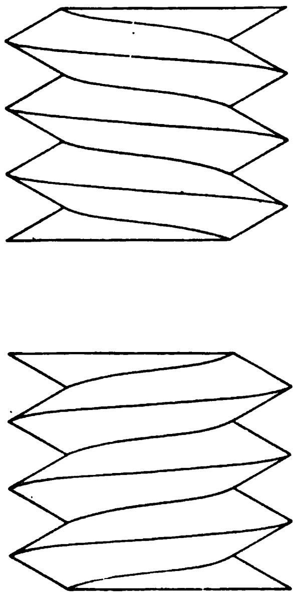


Fig. 11. V-Thread.

Fig. 13 shows the method of drawing a double V-thread. The process is exactly the same as for drawing a single thread. Start at point A, and draw the single thread A B C D exactly as in Fig. 12; then start at point E, half way between A and C, and draw another single thread of the same pitch as the first one. Some thought may be necessary to decide when the lines of one thread become hidden behind the other thread.

Square Thread. Another very common form of screw thread is that shown in Fig. 14, and known as the *square thread.* The

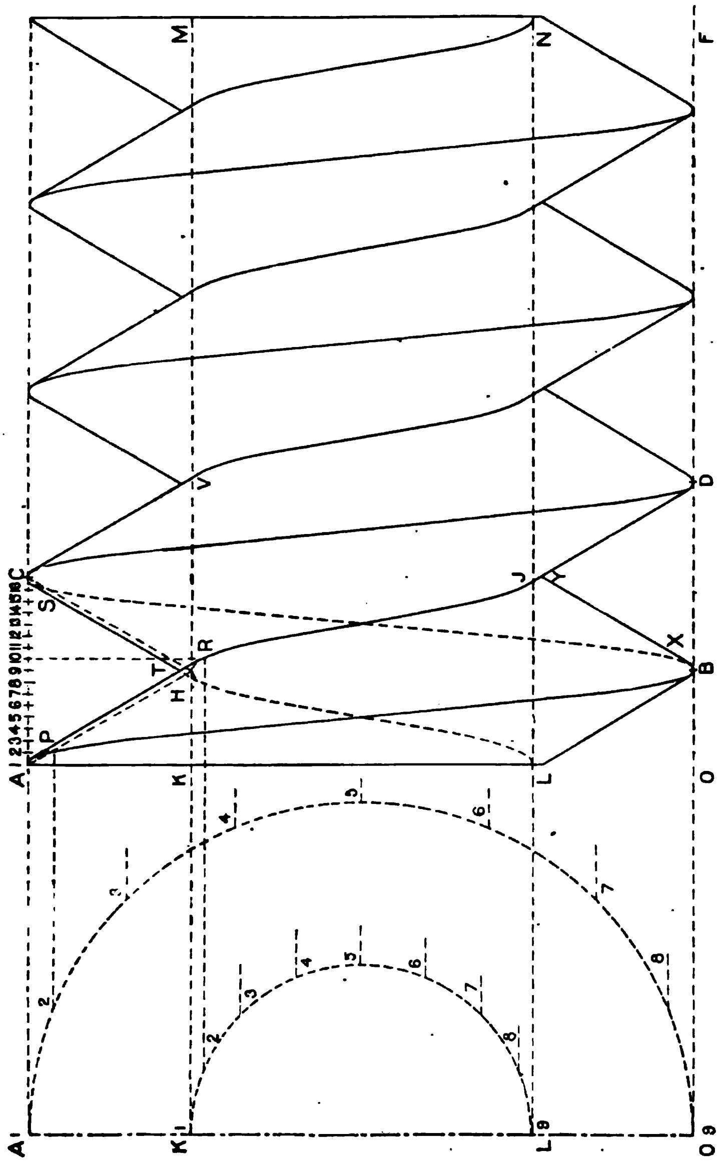


Fig. 12. Projections of V-Thread.

method of drawing this thread is similar to that for the V-thread,

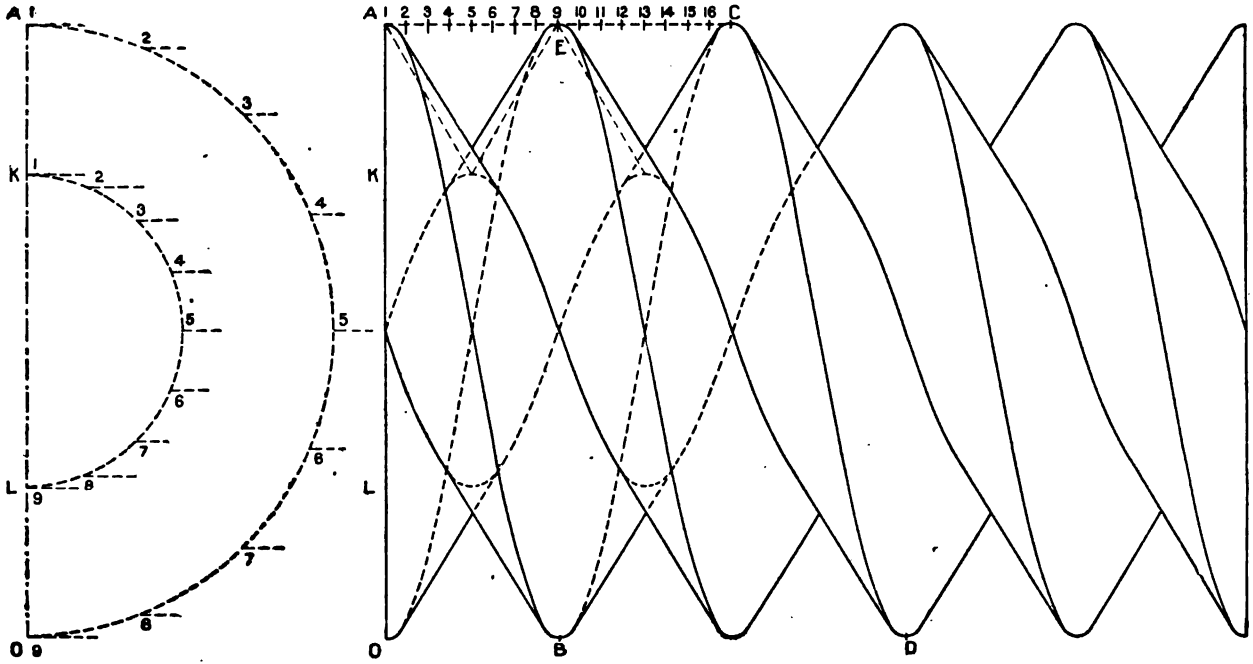
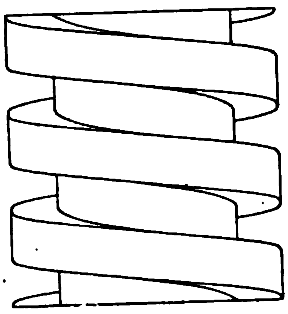
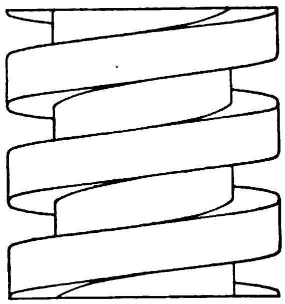


Fig. 13. Projections of Double V-Thread.

with the exception of a few minor points. The construction is shown in Fig. 15. The dimensions which must be known are the outside diameter A O, the pitch A C, the depth A H, and either the width of the thread A R, or the width of the groove R C. In the figure, the width of the thread A R is taken equal to one-half of the pitch; that is, A R and R C are equal. Beginning at A, draw the helix A B C; and beginning at R, draw the helix R M N, R N of course being equal to A C. Since the part between A and R is metal, forming the thread, there will be a line from A to R and from B to M, etc. Now, starting at point H, vertically under A, and at a distance from A equal to the depth of the thread, draw the helix H J V; and from S, vertically under R, draw helix S T W. Draw the lines S V, T N, etc. Here, as in the case of the V-thread, the invisible lines must be drawn when making the drawing, but need not be inked.



Left-hand.



Right-hand.

Fig. 14. Square Thread.

Fig. 16 shows the construction of a double square thread. An explanation is not necessary, since the difference between this and the single square thread is practically the same as between the single and double V-thread.

The V- and square threads are the two fundamental forms of thread in use, and all other forms are modifications of one or the other of these two. Figs. 17 to 20 show some of the more common modifications.

Figs. 17 and 18 show the two forms of the V-thread which are commonly used in practice.

In Fig. 17 we have what is known as the *Sellers* or *United States standard thread*, an enlarged drawing of which is shown in Fig. 21. Referring to this figure, we see that the angle between the two sides of the thread is 60 degrees, so that if the thread came to a point at the top and bottom, as indicated by the dotted lines, the depth of

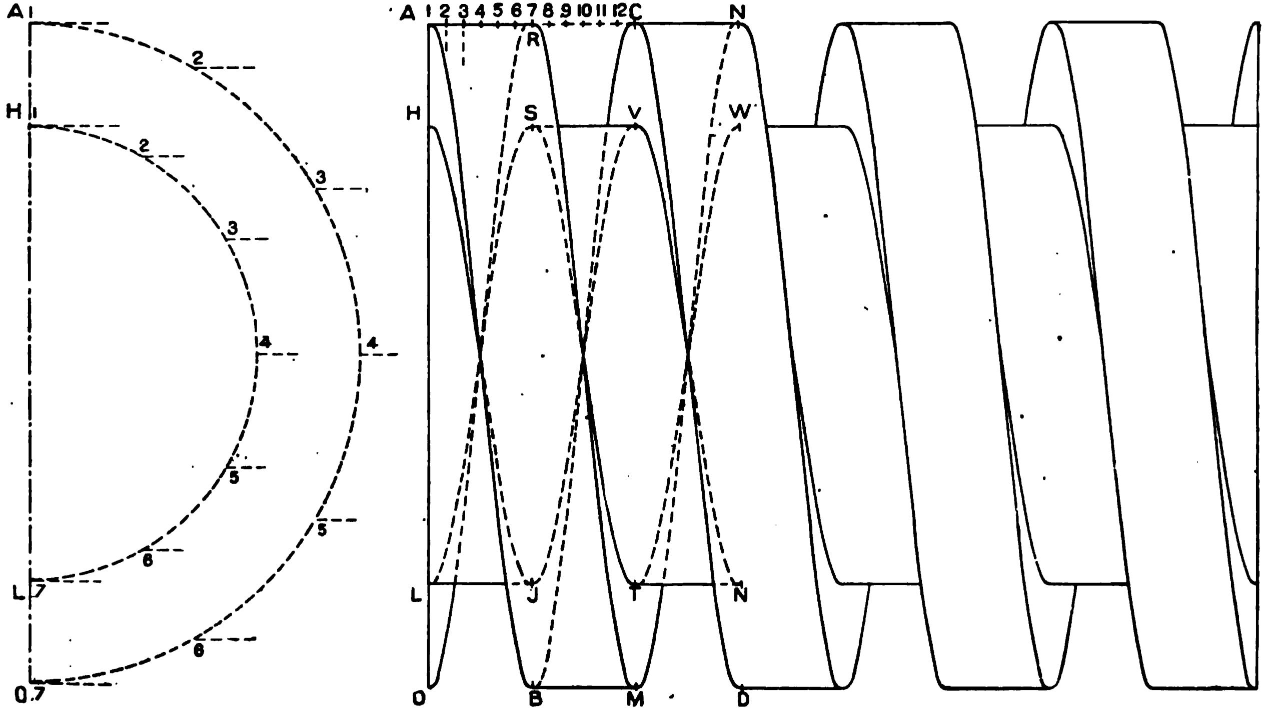


Fig. 15. Projections of Square Thread.

the thread D would be about $\frac{87}{100}$ of the pitch P. The sharp corners however, are a disadvantage, since on the outside they are likely

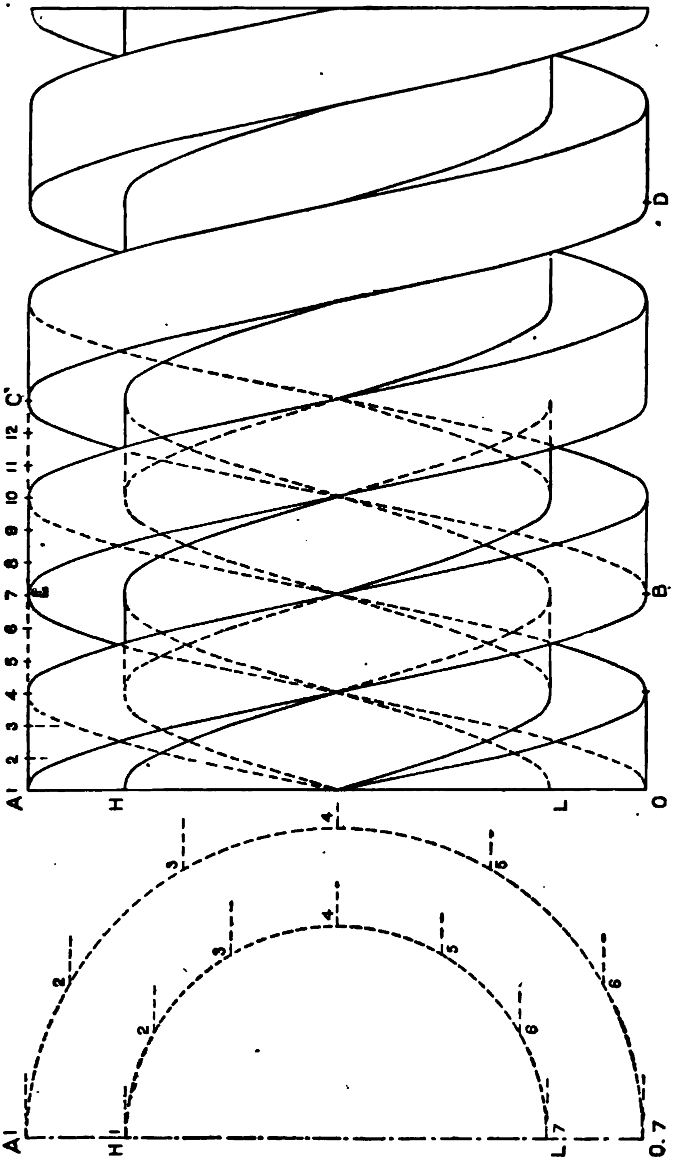


Fig. 16. Projections of Double Square Thread.

to be bruised and to give trouble in putting on the nut, and at the

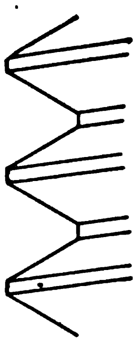


Fig. 17. U. S. Standard V-Thread.

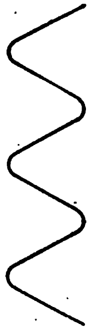


Fig. 18. Whitworth Standard Thread.

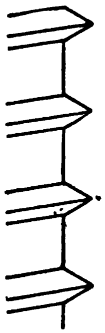


Fig. 19. V-Thread used on Wood Screws.

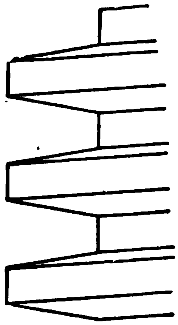


Fig. 20. Square Thread.

bottom of the groove they tend to weaken the bolt or screw. In order to avoid these sharp corners, the threads are flattened in the United States standard thread, as shown in Fig. 21, the amount of this flattening being such that the distance C is $\frac{1}{8}$ of the pitch, or—what amounts to the same thing—the distance A is $\frac{1}{8}$ of D. This gives a thread whose depth E is $\frac{65}{100}$ of the pitch.

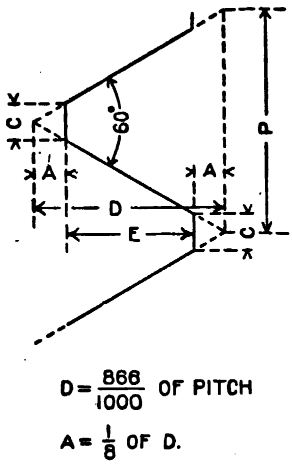


Fig. 21. Detail of U. S. Standard V-Thread.

Fig. 18 illustrates what is known as the *Whitworth standard thread,* shown enlarged in Fig. 22. Here the angle between the sides of the thread is 55 degrees, so that if the threads came to sharp corners, as shown by the dotted lines, the depth D would be $\frac{96}{100}$ of the pitch. The top and bottom of the thread, instead of being flattened, are rounded off so that the distance A is $\frac{1}{6}$ of D, or the depth E is $\frac{64}{100}$ of the pitch.

Fig. 19 shows the V-thread as used on lag screws and other wood screws. Here the groove is much larger than the thread, because the wood into which it is to screw is weaker than the iron of which the screw is made.

Fig. 20 shows a slightly modified form of square thread, the only difference between this and the square thread previously described being that the sides of the groove taper slightly.

Conventional Representations. The student should understand the drawing of the threads, as previously explained; and every draftsman should be able to draw the true projection of a thread if he should have occasion to do so. It is evident, however, that the process is complicated, and on a screw of small diameter and pitch it would be difficult to follow out the construction. To avoid this labor the construction shown in Figs. 23, 24, and 25 may be adopted, straight lines being substituted for the projections of the helix. Fig. 23 shows the conventional representation of the plain, single, right-hand V-thread, the true projection of which was shown in Fig. 12. To make the conventional drawing (Fig. 23), draw the parallel lines A B and C D at a distance apart equal to the outside diameter of the screw, and draw the line A C perpendicular to these two lines. Along A, lay off the distances A E, E F, etc., each equal to the pitch. Along C D, lay off C H equal to ½ the pitch; and from H, lay off H I, I J, etc., equal to the pitch. Draw lines from A to H, E to I, etc. Now, if the depth of the thread A S is known, draw the lines S T and U V; and beginning at L, perpendicularly under a point half way between A and E, lay off L M, M N, etc., equal to the pitch. In like manner find the points O, P, R, etc., and draw the lines L O, M P, etc.; also A L, L E, H V, H O, etc. The dotted lines should be left out in the finished drawing, but are put in the figure to show the construction.

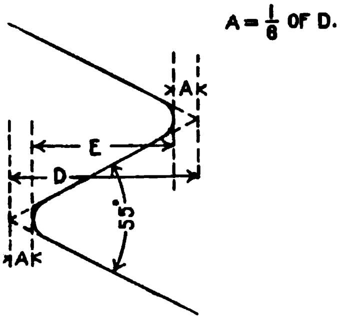


Fig. 22. Detail of Whitworth Standard Thread.

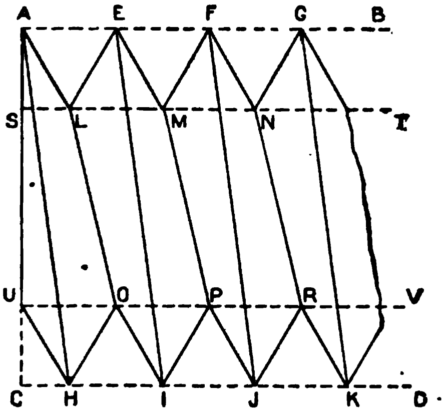


Fig. 23. Conventional Representation of Single, Right-Hand V-Thread.

If, instead of knowing the depth A S, we know the angle between A L and L E, the depth can be found by drawing from A and E the two lines A L and E L in such a way that they make the required angle with each other. To do this, the lines A L and E L should each make an angle with the line A B equal to 90 degrees minus $\frac{1}{2}$ the angle between A L and L E.

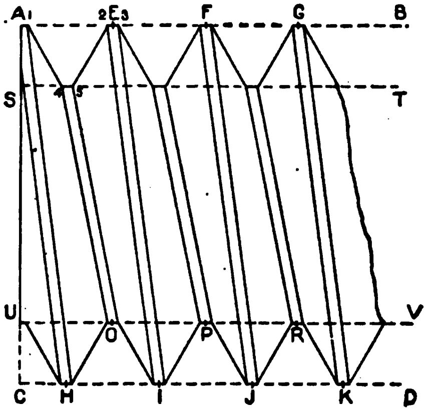


Fig. 24. Conventional Representation of U. S. Standard Thread.

Fig. 24 shows the corresponding construction for the United States standard thread. Draw the lines A B, C D, and A C as in Fig. 23, and find the points E, F, G, H, I, J, K, etc., in the same way as in that case. Now draw the lines S T and U V so that A S and C U shall equal $\frac{65}{100}$ of the pitch A E. On the line A B, lay off from A a distance A 1 equal to $\frac{1}{16}$ of the pitch; and each side of E, F, G, H, etc., lay off E 2, E 3, etc., each equal to $\frac{1}{16}$ of the pitch. From the points thus found, draw lines 1-4, 2-5, etc., making an angle of 60 degrees with A B. The rest of the drawing is completed as shown by drawing in full lines those parts of the lines A B, C D, S T, and U V intercepted between 1–4 and 2–5, etc.

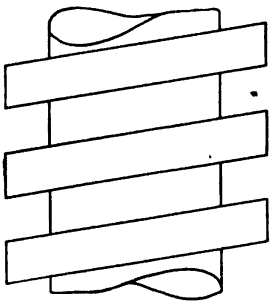


Fig. 25. Conventional Representation of Square Thread.

Fig. 25 shows the conventional representation of a square thread, and is drawn in exactly the same way as the true projection shown in Fig. 15, except that straight lines are used instead of curves, and certain other minor lines omitted.

Square threads are seldom conventionalized more than as shown in Fig. 25, and V-threads of coarse pitch and large diameter are usually drawn as in Fig. 23, whether sharp or U. S. standard. But for ordinary screws of small diameter and fine pitch, as are most frequently used, such a method involves too much labor, and the use of alternate long and short dashes across the body of the screw, as shown in Part IV, Fig. 14, is universally employed.

CAMS

A *cam* is an applied form of the ordinary wedge. The simple wedge is used to split apart the piece into which it enters, or to pry up heavy weights. It does not automatically repeat its work. The work of the wedge is finished when it is once driven home; and its function is not to produce motion, but to give mechanical advantage to the blows which drive it.

If we take a simple wedge and fasten it to some piece—say, a disc centered on a shaft, which is capable of continuous or periodic rotation, and allow the face of the wedge to rub against another guided piece, called the *follower*, we have a cam (Fig. 26). When we rotate the cam, it "wedges" the follower along a fixed path. When the follower reaches the top of the wedge, it may drop back to its original position by gravity or by the force of a spring, or it may be eased back by another wedge in the reverse position of the other. This cycle will be repeated as long as we choose to rotate the cam. The cam, therefore, is essentially a *repeating wedge,* and its function is primarily one of motion rather than great force. The wedge principle, however, enables very powerful cams to be made in cases where but little motion is desired. The motion of the cam is usually a rotation, but it may be an oscillation, or a straight-line reciprocating motion.

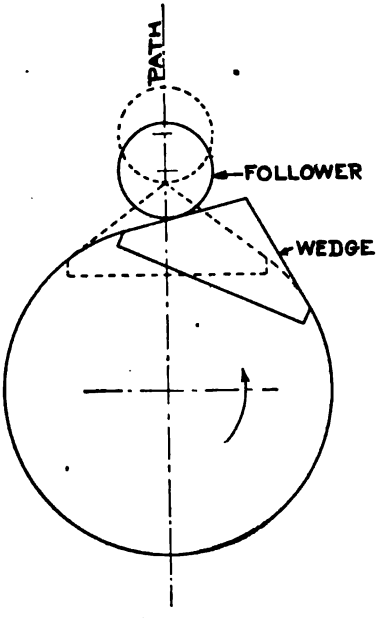


Fig. 26. Simplest Form of Cam.

In designing a cam, it is not only essential that the proper *lay-out* be made to produce, theoretically, the required *motion* of the follower, but that the wedge action be such that the cam will drive easily. A thin wedge can easily be driven under a heavy weight, whereas a thick, steep wedge cannot be made to enter. For precisely the same reasons, cams may be designed so steep in their wedge action that they drive with difficulty, and may even lock the follower in its path, on account of excessive side pressure. This point will be brought out more clearly in the discussions of the "Line of Pressure."

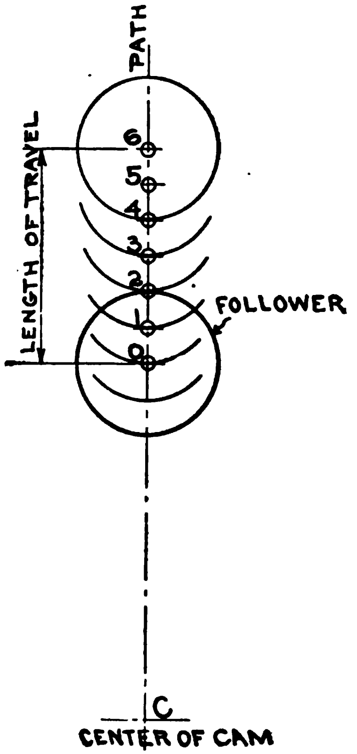


Fig. 27. Laying-out of Follower.

The actual laying-out of a cam is simple in principle, although somewhat tedious, especially in complicated cams. Several positions of the follower in its path are chosen, the follower drawn in those positions, and then the face of the cam is drawn tangent thereto. In order to do this, the several positions of the follower in its path may be laid down on the drawing paper (Fig. 27), and the cam itself developed on a separate piece of transparent paper or tracing cloth (Fig. 28). A pin is put through the center of rotation of the cam marked on the tracing cloth; and as the cam is brought to its several positions, the follower is traced in from its corresponding position on the paper below. When the movement of the cam is complete we have on the cloth a series of drawings of the follower; and if we draw a tangent line to these, the line of the cam will be produced. This method, however, from the drawing-board standpoint, is clumsy and inaccurate, because of the wearing of the pinhole and the error of transferring the shape of the follower to the tracing cloth. It is readily seen by reference to Figs. 27 and 28, that the same result will be attained if we artificially *rotate the follower about the cam* on the drawing-board by means of a pair of compasses. This is shown in Fig. 31, and is the method usually followed in cam design. The tracing-cloth method is useful in studying complicated cam movements; and also in testing the cam development when it is completed, to make sure that no error has been made.

From the above principles the uniform course of procedure in laying out a cam is as follows:

1. Draw the follower in several positions in its path.
2. Draw cam radii corresponding to these positions.
3. Rotate the follower about the cam.
4. Draw tangent line.
5. Test the cam.
6. Draw the line of pressure for each position.

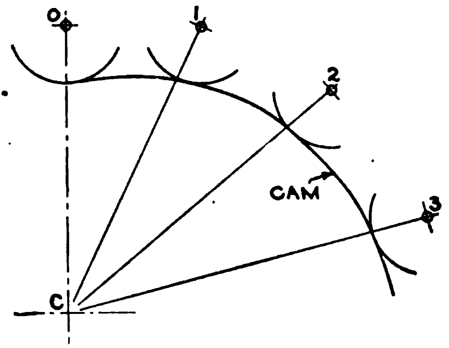


Fig. 28. Development of Cam.

Plate Cams. In Figs. 29 to 36 inclusive, it is required to move the follower F with uniform motion from position O to 6, while the cam rotates through 150°; it is then to remain at rest during a cam movement of 30°; then to return to its original position, moving uniformly from position 6 to O, while the cam turns through the remainder of the circle, or 180°. This is briefly expressed as follows:

Uniform rise	150°.
Rest	30°.
Uniform fall	180°.

PRACTICAL EXAMPLES

Fig. 29.—1. *Follower Positions.* The length of travel O 6 is supposed to be known, being fixed by some requirement of the machine to which the cam is to be applied. This distance, for a uniform motion, should be divided into any convenient number of equal parts; the more divisions, the more accurately can the cam be drawn. In this case six spaces are chosen.

2. *Cam Radii.* The diameter of base circle D is arbitrary; and its center having been chosen, draw the original radius C O; then the radii C X and C B, limiting the arcs of rise, rest, and fall respectively, should be drawn. As the follower must rise $\frac{1}{6}$ of its

travel while the cam rotates $\frac{1}{6}$ of its arc, there must be as many equal divisions of the cam arc as there are of the follower travel. Hence the arc of rise O X is divided into six equal parts and the radii drawn. Similarly, the arc of fall O B is divided into six equal parts and the radii produced.

3. *Follower Rotation.* The rotation of the follower about the cam is accomplished by setting the point of the compasses at **C,** and, with radius C 1, striking an arc intercepting the radius corresponding to position of the follower at R_1. Similarly points R_2, R_3, R_4, R_5, and R_6 are found. As the follower rests from X to B, the arc of intersection for R_6 is continued to F_6. For the period of fall,

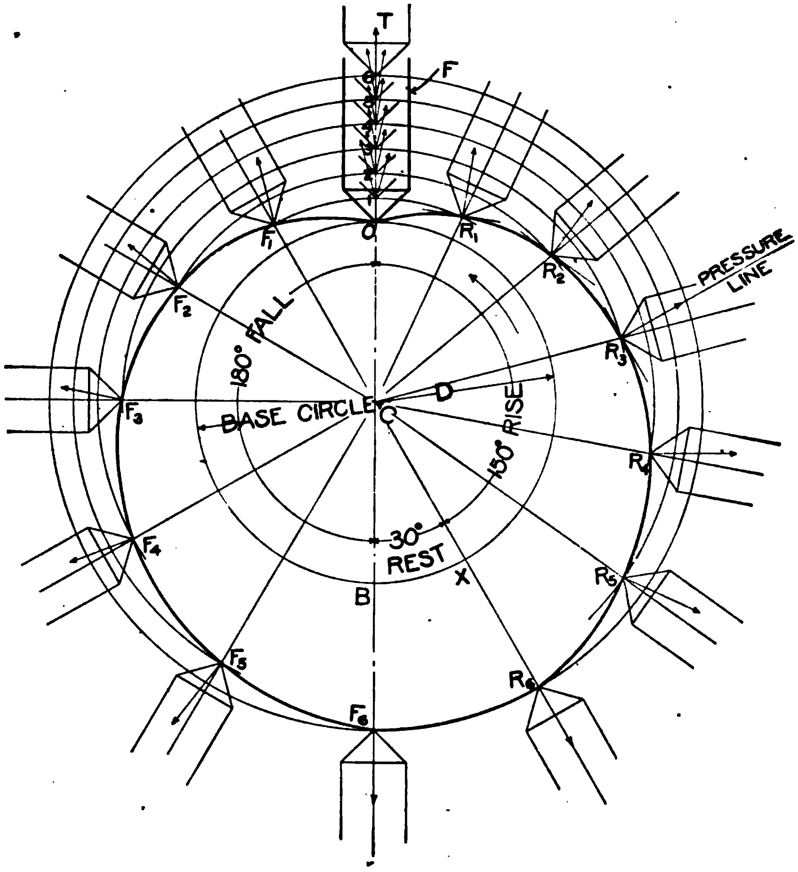


Fig. 29. Pointed Follower with Path Intersecting Center of Cam.

arcs are swung from the same points of follower travel as before, making the intersections F_5, F_4, F_3, F_2, F_1.

4. *Tangent Line.* A smooth curve is now drawn through the points of intersection, thus forming the outline of the cam. For other forms of follower than a sharp point, this line would be strictly a tangent line to the face of the follower. A pointed follower, as shown in the figure, is not a very practical form, as the point is subjected to severe wear. It is chosen for the present illustration, to afford the simplest possible cam development.

5. *Testing.* The cam may be tested by laying over it a piece of tracing cloth, and tracing roughly the outline of the cam, also marking the radii and the center of rotation. A pin is now placed at the center of rotation of the cam, and the tracing cloth swung until the several radii C R_1, C R_2, C R_3, etc., fall into the line of travel C T. If the cam is correctly developed, it will just touch the several positions of the follower in its path when the proper radius is coincident with the line of travel C T. It is very easy to make a mistake in laying out cams, especially the more complicated ones; and this rough method of proving the work should always be applied.

6. *Pressure Line.* The face of the follower in this case is a point, and the pressure line, being the common normal between the cam and follower at point of contact, is always *theoretically* normal to the cam at that point.

During motion, however, the force of friction between the cam and follower would modify somewhat the direction of the pressure line, turning it so as to produce a side thrust against the follower, causing consequent chattering and possible binding in its guides. This can be minimized by ample lubrication and hardened faces; but for cams which have any considerable load to work against, a follower carrying a roll against the cam is a necessity.

The line work in cam design should be fine and accurate. A hard pencil, kept well sharpened, is necessary, and special care must be taken to get definite intersections. In order to keep the center of the cam in as good condition as possible, it is well not to continue the radii to the center, but to stop when a short distance from the center, as shown. When penciling and inking in, use a fine, continuous line, not dotted; the continuous line is more quickly made and is apt to be more accurate than the dotted line. Moreover, cam is strictly *lay-out* work, not finished in detail, and the subsequent detail drawing of the cam should not be confused with the lay-out of the cam *outline*.

Fig. 30.—1. *Follower Positions.* The follower positions are chosen and drawn precisely as in Fig. 29; in this case, however, the path of the follower does not intersect the center of the cam, but if produced would pass at some distance to one side of it. This changes materially the development of the cam, as will subsequently be noted.

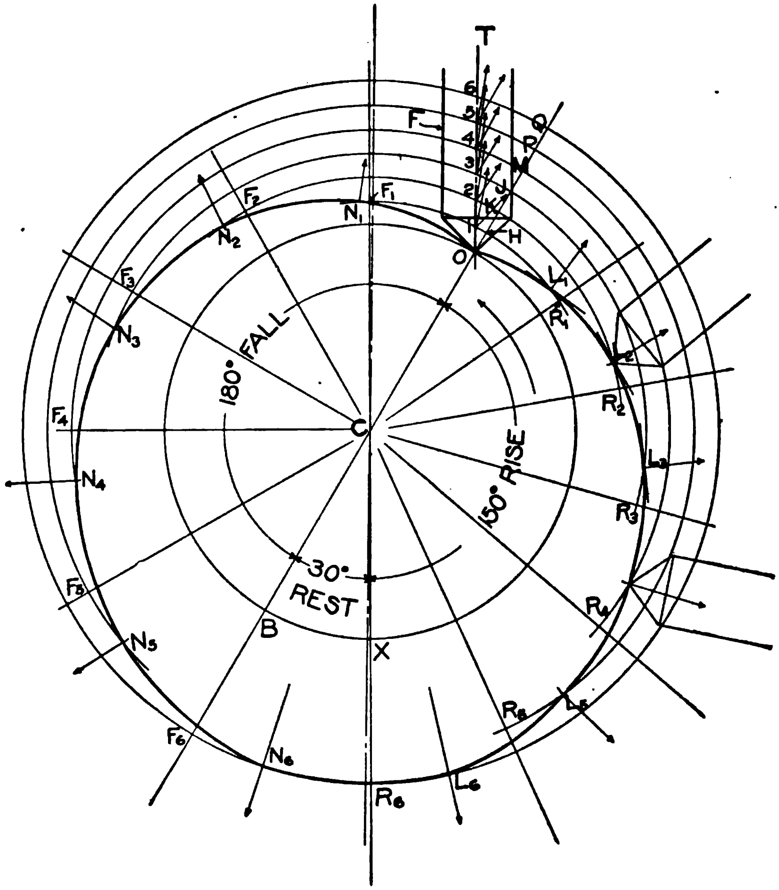


Fig. 30. Pointed Follower with Path not Intersecting Center of Cam.

2. *Cam Radii.* The diameter of base circle and center of cam being chosen as before, draw the original radius C O. This original

radius is the one to which all subsequent radii are related. Treating this radius the same as the original radius C O in Fig. 29, the radius C X is drawn, making the arc of rise 150° as before. The radius C B is then drawn, giving the 30° arc of rest, and leaving the arc of fall between C B and C O 180° as before. Since the cam is to have uniform motion, the subdivisions of the arcs are equal, and are made in exactly the same manner as in Fig. 29.

3. *Follower Rotation.* The follower is now rotated about the cam by the compasses, in similar manner to that in Fig. 29, producing the intersections R_1, R_2, R_3, R_4, etc. Now, however, the introduction of a new step should be noted. The original radius C O, not being coincident with the path of the follower O T, the intersections R_1, R_2, R_3, etc., do not represent the actual positions of the point of the follower when rotated. During the arc of rise, the follower is gaining on the radius, and the increasing distances R_1 L_1, R_2 L_2, R_3 L_3, etc., must be set off ahead of the radii, equal to the distances H 1, K 2, J 3, etc. In similar fashion, during the arc of fall, the decreasing distances F_6 N_6, F_5 N_5, F_4 N_4, F_3 N_3, must be set off ahead of the radii, equal to Q 6, P 5, M 4, J 3, etc.

4. *Tangent Line.* A smooth curve is now drawn, not through the original points of intersection with the cam radii, as in Fig. 29, but through the points *set off as above from these intersections.* In other words, the cam curve is drawn through points L_1, L_2, L_3, L_4, etc., and N_6, N_5, N_4, N_3, etc.

5. *Testing.* A piece of tracing cloth should be laid over the cam, the outline traced upon it, the radii marked, and then the tracing cloth rotated about the pin point, as in Fig. 29, to see if the cam in its successive positions just touches the follower in each of its positions.

6. *Pressure Line.* The pressure line at each position of the follower is, as in Fig. 29, normal at the point of contact.

Fig. 31. 1. *Follower Positions.* In Fig. 31, the case is identical with that of Fig. 29, except that the shape of the follower has been changed to the more practical form of a roll, which can turn about a pin, thus relieving the crowding, grinding action characteristic of the pointed follower hitherto discussed. The path of follower roll F is divided as before, into six equal parts.

2. *Cam Radii.* The original radius C O, the radius C X, limiting the arc of rise, the radius C B, limiting the arcs of rest and fall, are all drawn precisely as before, and the subdivisions of the arcs are in nowise changed from the preceding cases.

3. *Follower Rotation.* The follower is now rotated about the cam, giving the intersections R_1, R_2, R_3, R_4, R_5, and R_6 for the period

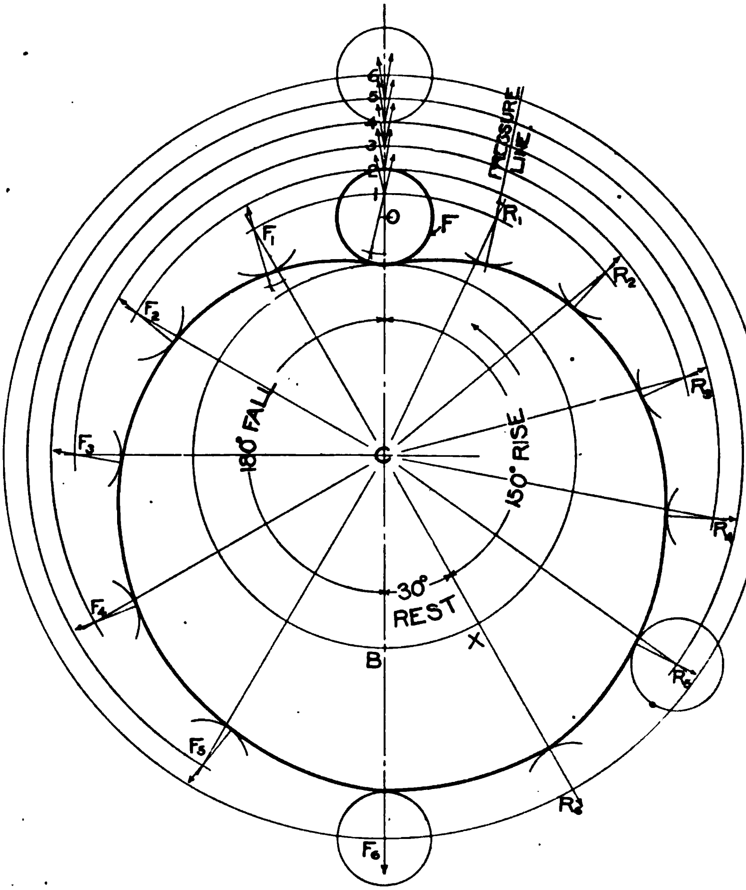


Fig. 31. Follower in Form of Roll with Path Intersecting Center of Cam.

of rise; the corresponding intersections for the period of fall are F_6, F_5, F_4, F_3, F_2, and F_1. With each of these intersections as a center, and a radius equal to the radius of the follower roll, an arc is struck, which represents the follower in its rotated position.

4. *Tangent Line.* A common tangent line is now drawn to the several positions of the rotated follower, giving the outline of the cam as a smooth curve. In order to give the follower its full period of rest from R_6 to F_6, the portion of the cam lying between the radii C X and C B must be a true arc of a circle struck from center C. Special attention must be paid to this point, because, if the true arc is not maintained between these radii, the full period of rest will not be secured.

5. *Testing.* The testing of the cam is accomplished in the same way as previously described, by the tracing-cloth method. The several positions of the follower in its path should be drawn; and as the cam is rotated into its several positions, if the work has been accurately done, the cam will be perfectly tangent to each position of the follower.

6. *Pressure Line.* By a well-known principle of mechanics, when two bodies are in contact, the line of pressure between them is always perpendicular to the surfaces at their point of contact; in other words, the line of pressure is the common normal at the point of contact. The follower being a true circle, the perpendicular at any point of its surface must pass through the center of the roll; therefore, if we draw a line from the center of each position of the roll to the point of contact between it and the cam, this line will be the line of pressure between the cam and the roll. This has been done in Fig. 31, and the arrows indicate the direction of the pressure of the cam against the roll. In order to group these lines of pressure so that the action may be clear as the follower moves over its path, the lines of pressure as drawn are rotated back to the corresponding points in the path of the follower. By this it is readily seen that during the period of rise the lines of pressure are all slightly inclined towards the left of the line of travel, while during the period of fall the lines of pressure are all slightly inclined towards the right. The cam as shown in Fig. 31 is a very good cam, as far as the lines of pressure are concerned. The ideal condition would be to have the lines of pressure all coincident with the line of travel. This is impossible, because the only shapes which would give a common normal along the line of travel would be two circles, and a cam with a perfectly circular outline could, of course, give no travel to the follower The fact that the lines of pressure are at such a slight angle to the line of

travel indicates that there is very little side pressure on the follower and that, therefore, the cam will be an easy working cam.

Fig. 32. 1. *Follower Positions.* The follower positions are chosen precisely as in Fig. 31, and the subdivisions of the path of the follower similarly made. This case corresponds with that of Fig. 30 the line of travel not intersecting the center of the cam.

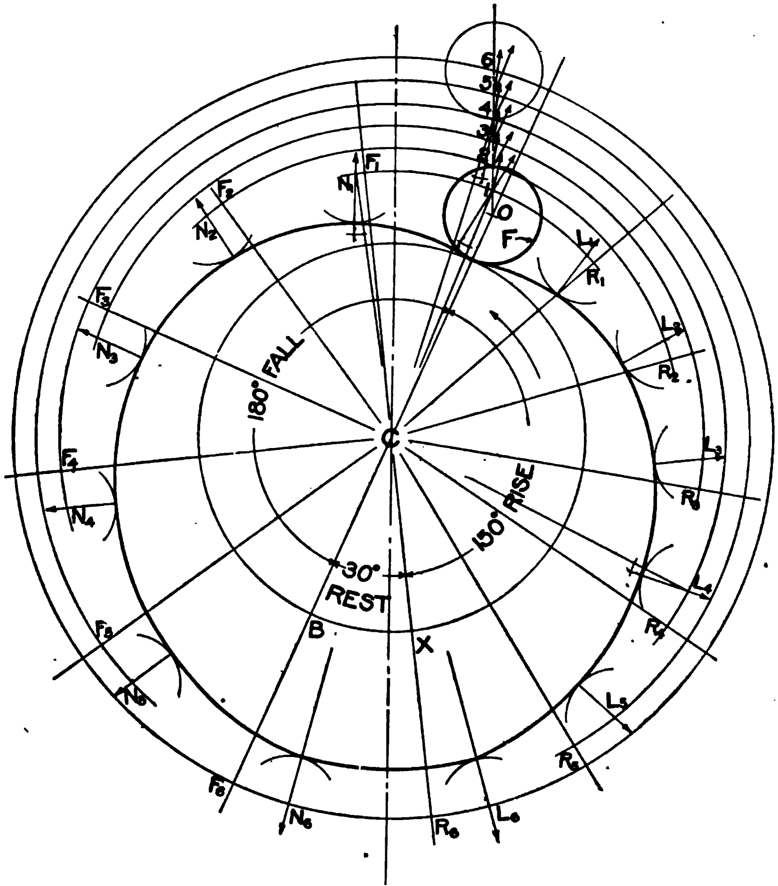


Fig. 32. Follower in Form of Roll with Path not Intersecting Center of Cam.

The shape of the follower, however, is a roll similar to that just discussed in Fig. 31.

2. *Cam Radii.* The original radius C O is drawn as before. Then the radius C X, limiting the arc of rise, the radius C B, limiting

the arcs of rest and fall, and the subdivisions of the arcs of rise and fall, are made exactly as in Fig. 31.

3. *Follower Rotation.* . The treatment of follower rotation is the same as that in Fig. 30. The intersections R_1, R_2, R_3, etc., being found, the distances R_1 L_1, R_2 L_2, R_3 L_3, which the follower gets ahead of the radii, are set off exactly as in Fig. 30. In this case, however, the points L_1, L_2, and L_3 are the centers of the rotated follower roll; and from these centers are struck the arcs representing the follower roll in its several rotated positions.

4. *Tangent Line.* A smooth tangent line is now drawn to the several positions of the rotated follower, thus giving the outline of the cam. In this case the outline of the cam giving the period of rest lies between the radii C N_6 and C L_6. In order that the full period of rest may be accomplished, it is necessary that the portion of the cam between these two radii be a true arc, struck from center C. Special attention should be given this point.

5. *Testing.* The cam should be tested by the tracing-cloth method, as before.

6. *Pressure Line.* The pressure lines are drawn as in Fig. 31 by joining the center of the follower roll in its rotated positions to the point of contact between the cam and the roll.

Arrows, as before, indicate the direction of the pressure. These pressure lines may be rotated until they are collected along the line of travel, indicating the change in direction of the pressure between the cam and the roll as the roll passes over its path. A convenient method of laying off these pressure lines is to join the center of the cam C with the center of each roll; measure the angle which the line of pressure makes with this radius; and then transfer the angle to the proper point on the line of travel. This method is clearly indicated in the figure.

As before, it will be seen that the line of pressure lies quite close to the line of travel, and therefore the cam will be an easy working cam.

Fig. 33.—1. *Follower Positions.* In this case the follower is a roll, as before; but instead of traveling in a straight line, it is made to travel along the arc of a circle, being carried on the end of an arm O C′, C′ being a fixed point about which the arm oscillates. The length of travel O 6 is the same as before, and is likewise divided

into six equal parts. This method of carrying the follower roll is perhaps the most common of all, and is a very effective plan for giving the follower roll easy movement along its path.

2. *Cam Radii.* The original radius C O, is drawn in this case, as in all the others, through the original center O of the follower roll. The radii C X and C B limiting the arcs of rise, rest,

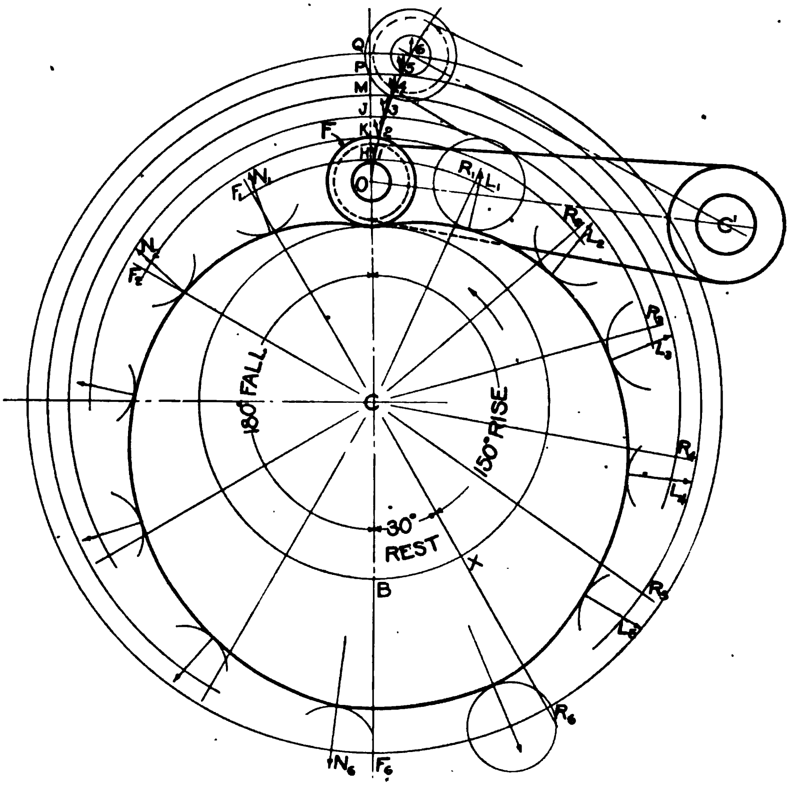


Fig. 33. Follower in Form of Roll Mounted on Oscillating Arm.

and fall are likewise drawn in the given relation to the original radius C O, and the arcs subdivided precisely as before.

3. *Follower Rotation.* For the purpose of follower rotation, arcs are now struck through the points 1, 2, 3, 4, 5, and 6, these arcs being prolonged until they meet the original radius in the points H, K, J, M, P, and Q. Then the rotation of the points H, J, K, etc., produces the intersections R_1, R_2, R_3, etc.; but it should be noted

in this case that the follower roll, instead of getting ahead of the radius, as in Figs. 30 and 32, is lagging behind it at each position. The distances R_1 L_1, R_2 L_2, R_3 L_3, R_4 L_4, R_5 L_5, while being laid off equal to H 1, K 2, J 3, M 4, P 5, Q 6, etc., as in Figs. 30 and 32, are laid off behind the radius in each position on the arcs of rise and fall. These distances are constantly increasing up to point 6, where the roll remains stationary during the period of rest, and then constantly decrease to zero, until the roll reaches the original position at point O. From the points just found, arcs are struck as before, the radius being equal to the radius of the follower roll.

4. *Tangent Line.* The tangent line is drawn as a smooth curve to these arcs, and the arc of rest is struck as before, thus developing the outline of the cam.

5. *Testing.* The cams should be tested by the tracing-cloth method as before.

6. *Pressure Line.* The pressure lines are drawn precisely as in Figs. 31 and 32; but it is a little more difficult to rotate these pressure lines back to the points in the path of the follower, and the tracing-cloth method is suggested as best for this purpose. This is done by taking a scrap piece of tracing cloth, fixing a pin through it to the center of the cam, tracing upon it, from the paper below, the pressure lines and the centers of the follower. These centers being rotated back until coincident with the corresponding points of the travel, a second point in each pressure line is pricked through on the paper below. Upon the removal of the tracing cloth each pressure line can then be quickly drawn through these pricked points and the corresponding centers of the follower, thus enabling the action of the cam to be properly judged.

Fig. 34.—In this case the follower roll has been abandoned, and the original pointed follower substituted. The motion of the follower point, however, instead of being in a straight line, is in the arc of a circle precisely as in Fig. 33, except that the follower, being a point instead of a roll, the points L_1, L_2, L_3, etc., have the outline of the cam drawn directly through them. This case is introduced merely for the purpose of making the set of cams complete, and it is so closely a counterpart of Fig. 33 that the detail study of it is considered unnecessary.

Fig. 35.—1. *Follower Positions.* Here is introduced a follower with a flat surface, its path being perpendicular to its working face. The length of its path O 6 is the same as before, and is divided into six equal divisions.

2. *Cam Radii.* The point of original contact O being chosen, the original radius O C is drawn perpendicular to it; the radii C X and C B, limiting the arcs of rise, rest, and fall, are then drawn in their proper relation to C O, and the arcs of rise and fall subdivided as before.

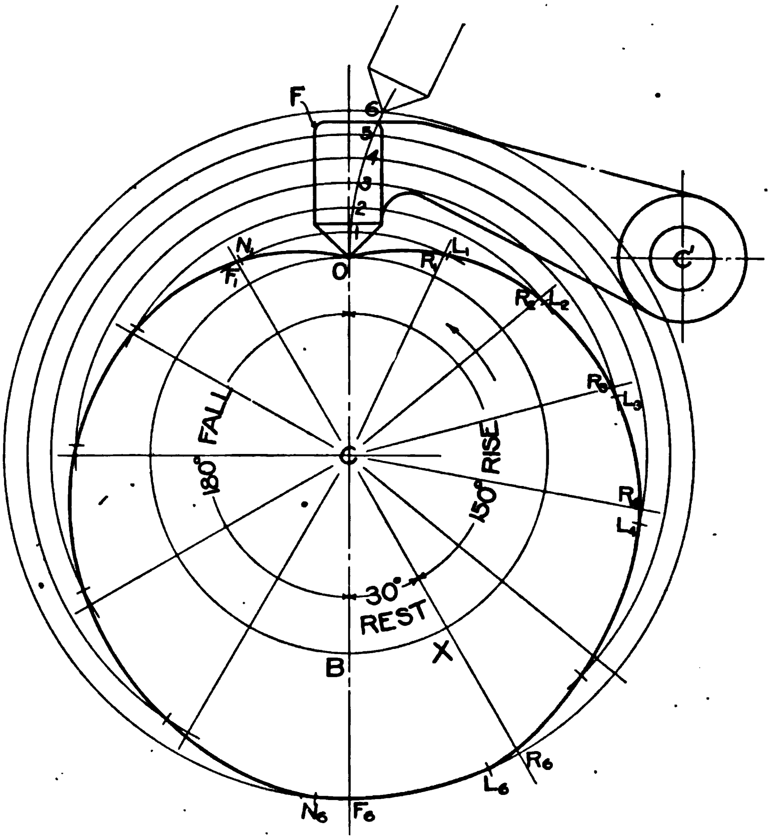


Fig. 34. Pointed Follower Mounted on Oscillating Arm.

3. *Follower Rotation.* The intersections R_1, R_2, R_3, R_4, etc., of the rotating arcs with the several positions of the radii, are found as before. The rotated positions of the follower in this case are

obviously represented by drawing perpendiculars to the several radii through the points R_1, R_2, R_3, R_4, etc.

4. *Tangent Lines.* The outline of the cam is produced by drawing a tangent line to the several lines representing the rotated positions of the follower, the arc of rest being struck as before.

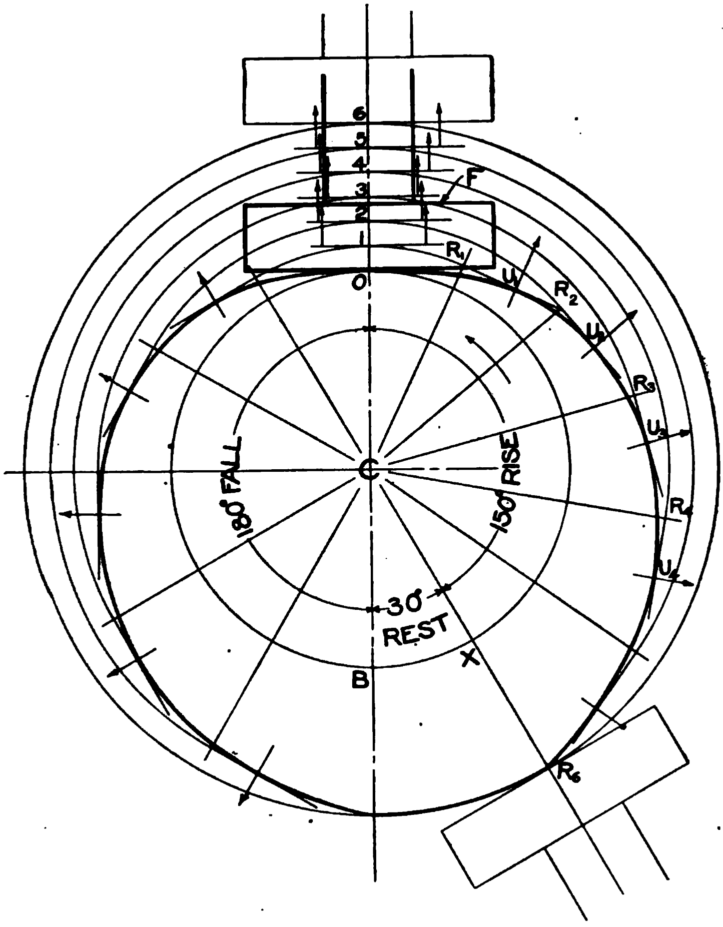


Fig. 35.- Follower with Flat Surface. Path Perpendicular to Working Face.

5. *Testing.* The cam should be tested by the tracing-cloth method.

6. *Pressure Line.* Pressure lines are drawn at the points of contact between the cam and the follower, by erecting perpendiculars

to the face of the follower at these points. As in the case of the pointed follower, there is considerable friction due to the sliding of the cam along the follower face. This friction produces a side thrust perpendicular to the path of the follower, and modifies the pressure lines slightly. If it were not for this friction, the pressure line obviously would always be perpendicular to the follower face, acting at a point on the follower face some distance to one side of the original point of contact O. By taking the distances R_1 U_1; R_2 U_2, R_3 U_3, etc., to the several contact points, and rotating them back, the manner in which the point of contact between the cam and the follower moves along the face of the follower during its travel can be conveniently studied; it is seen that the point of contact during the arc of rise moves to the right of the original radius, and gradually swings back again until, at the point 6, it is on the line of the original radius. During the arc of rest, the point of contact remains at point 6; during the arc of fall, it moves to the left of the original radius, finally coming back again to the original point of contact O.

Fig. 36.—1. *Follower Positions.* This is a flat-faced follower, carried by an oscillating arm similar to the roll in Fig. 33. The length of travel O 6 is divided into six equal parts, as in the previous cases, the positions of the follower being indicated by the radial lines C′ 1, C′ 2, C′ 3, etc.

2. *Cam Radii.* The original radius C O is drawn through the assumed point of contact, and perpendicular to the face of the follower in its original position. The radii C X and C B are then drawn limiting the arcs of rise, rest, and fall, and the subdivisions of the arcs of rise and fall properly made.

3. *Follower Rotation.* The intersections R_1, R_2, R_3, etc., of the rotating arcs, are found as in the previous cases. The rotation of the follower is accomplished by drawing through the points R_1, R_2, R_3, etc., straight lines making the same angle with these radii as the follower in its corresponding positions makes with the original radius C O. For example, the angle a_1 is equal to a; the angle b_1 is equal to b; and the angle c_2 is equal to c.

4. *Tangent Line.* The tangent line is now drawn to the several positions of the rotated follower, and the arc of rest is struck, thus giving the outline of the cam.

5. *Testing.* The cam should be tested by the tracing-cloth method.

6. *Pressure Line.* The pressure lines are drawn the same as in Fig. 35, and it is suggested that the rotation of them back to the positions of the follower in its path be accomplished by the tracing-cloth method, as in Fig. 33.

Remarks on Figures 29–36. It should be especially noted, that in all the cases of cams thus far studied, the methods of procedure are absolutely identical. In the more complicated cases of cams which follow, and others which may arise in the designing of special machines, the same principles apply, however much they may apparently be disguised by the conditions of the problem. A good method to follow in designing a complicated cam is first to study the cam by the tracing-cloth method, as described on page 18. After an *approximate* outline of the cam has been determined as being

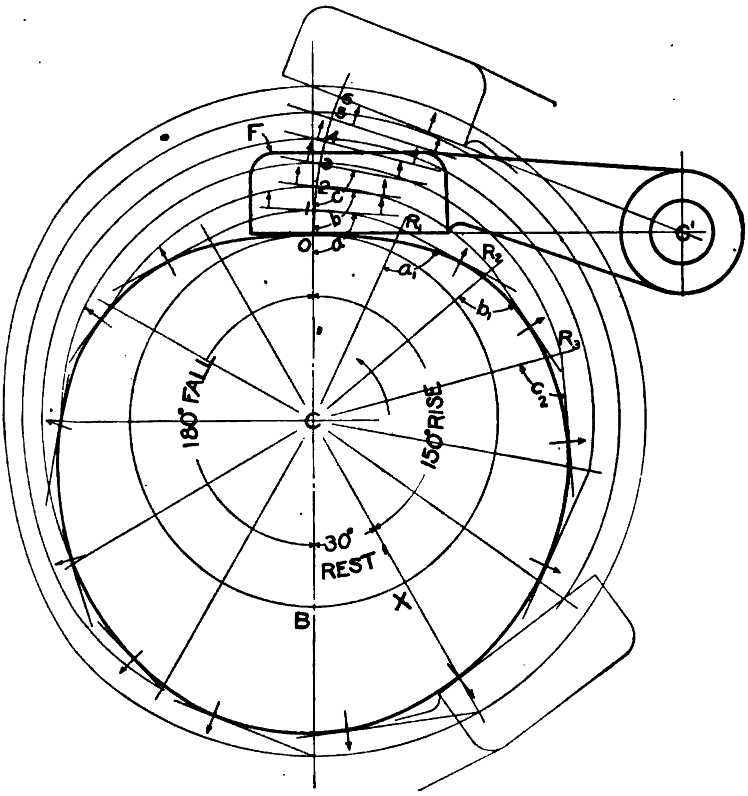


Fig. 36. Follower with Flat Surface Mounted on Oscillating Arm.

possible, and giving about the motion of the follower desired, then the more formal method of rotating the follower about the cam can be applied, to give the exact outline. It should be remembered that not all combinations of length of travel, angle of rest, rise, and fall, as fixed by the conditions of the problem, are possible to reproduce by cam movements. In such cases the cam is designed to give the nearest possible approach to the motion desired, or some of the conditions changed.

In each of the cases considered, it is assumed that the follower is always held against the surface of the cam, either by its own weight or by a spring. Another method of accomplishing this is to make a cam with two surfaces, the follower running between them in the groove thus formed.

Uniform Motion. All the cams thus far considered are *uniform-motion cams*, or cams having *equal rise* in *equal time*. By this is meant that each fraction of the travel of the follower is made in the same time as each other similar fraction; and likewise, that each fraction of the rotation of the cam is made in the same time as that of any other fraction of its movement. In further explanation of the term "uniform motion," suppose a railway train to travel ten miles in twenty minutes, the speed of the train being the same during each minute; it would travel ½ mile in each minute, and would have a *uniform* speed or velocity of ½ mile per minute. Under such conditions the train would be moving at the full speed of ½ mile per minute, both at the beginning and at the end of the ten miles. If, however, it had exactly ten miles to travel and exactly twenty minutes to do it in, and must be at rest at the beginning and at the end of the given time, it could not gain its full speed in an instant or lose it in an instant, but must start and stop gradually. Therefore, during the first part and the last part of the time, it would be moving at a speed slower than ½ mile per minute, and must go faster than ½ mile per minute during the middle part of the run, to make up for the time lost in starting and stopping. Such motion would not be uniform. The more suddenly the train starts and stops, the more nearly uniform the main part of the travel may be, but the greater the shock when starting and stopping.

Suppose we have a piece to be moved one foot in ten seconds; if the motion is to be uniform, the piece would have a velocity of $\frac{1}{10}$

foot per second. If the piece is light and the mechanism which does the moving is sufficiently powerful, the piece may be made to start and stop almost instantly without serious shock, and consequently may have practically uniform motion.

Varying Motion. If the mechanism which moves the piece be so designed as to start and stop it gradually, the shock will be avoided. We may have a gradual increase of speed at the start, until full speed is attained; then a uniform full speed during the main part of the stroke; and finally a gradual decrease of speed to a full stop at the end of the stroke; or the speed may increase during the entire first half of the stroke, and decrease during the entire last half, the motion at no time being uniform. The first condition is used where the piece is doing work during the stroke, as, for instance, the cutting stroke of a tool; and the second condition, where quick motion is desired without regard to its character, so long as it is without shock.

Harmonic Motion. In Fig. 37, let the semicircle A B D represent the path along which a piece moves with a uniform velocity.

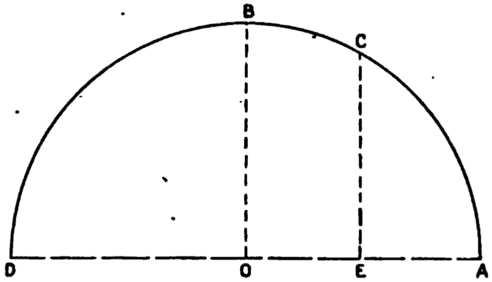


Fig. 37. Illustration of Harmonic Motion.

Now, if we have another piece moving along the diameter A D, starting from A at the same time as the first piece, and moving at such speed that a perpendicular let fall from any position of the first piece to the line A D will locate the second piece—that is, when the first piece is at C, the second piece is at E; when the first piece is at B, the second is at O; and so on—then the piece which travels along the line A D has *harmonic motion*. It moves slowly at first, increases to a maximum speed at the center, and decreases to rest at the end. This motion is quite common for shaper rams, slotters, and feed-mechanisms. Cams can readily be designed to give such a motion.

Uniformly Accelerated and Retarded Motion. A piece which has uniformly accelerated and uniformly retarded motion (see Mechanism) moves through one unit of space in the first unit of time, three the second, five the third, seven the fourth, nine the fifth, etc., to the middle of its stroke, then decreases at the same rate to the end of the stroke. For example, if a piece is to move with uniformly accelerated and retarded motion one foot in ten seconds, it will move $\frac{1}{50}$ foot the first second, $\frac{3}{50}$ the second, $\frac{5}{50}$ the third, $\frac{7}{50}$ the fourth, $\frac{9}{50}$ the fifth, when it will have traveled $\frac{1}{50} + \frac{3}{50} + \frac{5}{50} + \frac{7}{50} + \frac{9}{50}$ $(=\frac{25}{50})$, or $\frac{1}{2}$ the whole foot in one-half the whole time, the speed increasing all the time; at the end of the fifth second, when half the distance has been traveled, it begins to slow down, and travels $\frac{9}{50}$ the sixth second, $\frac{7}{50}$ the seventh, $\frac{5}{50}$ the eighth, $\frac{3}{50}$ the ninth, and $\frac{1}{50}$ the tenth. The rate at which the velocity increases during the first half of the time is often made the same as that at which the velocity of a weight increases when dropped from a height; and the rate at which the velocity decreases during the last half of the time, the same as that at which the velocity of a weight decreases if thrown straight up into the air. This particular form of uniformly accelerated and uniformly retarded motion is, therefore, known as *gravity motion*. It is commonly produced by cams, although not often used for motions greater than a few inches.

If in Fig. 29 it had been required to raise and lower the follower with *harmonic* instead of uniform motion, the only difference in procedure would have been in dividing up the path O 6. The divisions of O 6, instead of being equal, would be found as shown in Fig. 38. Suppose the line Ae to be the same length as O 6; on this line as a diameter, draw a semicircle, and divide this semicircle into as many equal parts as the arc of rise is divided. In this case, suppose the arc of rise to be divided into 8 equal parts; then the semicircle is likewise divided as shown in the figure. From the points B, C, D, etc., drop perpendiculars to the line Ae, cutting it at points 1, 2, 3, etc. For the first eighth of the arc of rise of the cam, let the follower rise the distance A-1, for the second eighth the distance 1-2, for the third eighth the distance 2-3, and so on. Such motion of the follower is harmonic motion.

If it is required that the follower shall rise and fall with *uniformly accelerated and retarded* motion, the method of dividing the line of travel Ae is shown in Fig. 39. As in the case of harmonic motion, suppose the arcs of rise and fall each to be divided into 8 equal parts. Now, the line Ae must be divided into 8 parts, but these parts must be such that, beginning with the point A, the distances A-1, 1-2, 2-3, and 3-4 shall be in the ratio of 1, 3, 5, and 7; and the distances 4-5, 5-6, 6-7, and 7-e shall be in the ratio of 7, 5, 3, and 1; in other words A-1 is $\frac{1}{32}$ of the whole line Ae, 1-2 is $\frac{3}{32}$ of Ae, 2-3 is $\frac{5}{32}$ of Ae, and so on. To divide up the given length Ae so that the divisions may bear the above relation to one another, draw the line Ar at any convenient angle, and, choosing any convenient distance as a unit, mark it off on this line 32 times, beginning at A. From I, the last of these dividing points, draw a line to e; next find the point B, the 1st division from A; C the 3rd from B, D the 5th from C, E the 7th from D, F the 7th from E, G the 5th from F, H the 3rd from G; and through the points H, G, F, E, etc., draw lines parallel to Ie, cutting the line Ae at 7, 6, 5, 4, etc. It is obvious that Ae is then divided in the same proportion as AI, and if the follower were made to travel along the line Ac according to these divisions, it would have uniformly accelerated and retarded motion.

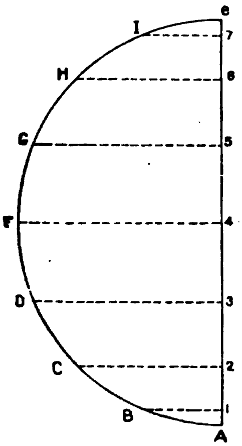


Fig. 38. Harmonic Motion.

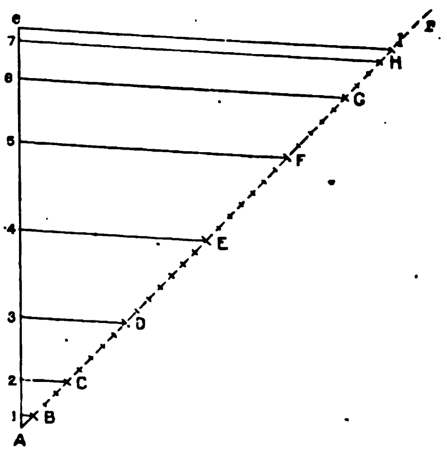


Fig. 39. Uniformly Accelerated and Retarded Motion.

EXAMPLES FOR PRACTICE

For the purpose of illustrating these principles, suppose it is required to design a plate cam, such that the follower rises from point A to e, Fig. 40, with harmonic motion, while the cam rotates through 120°; it is then to remain at rest during the cam movement of 60°; it is then to fall to its original position with a uniformly accelerated and retarded motion, while the cam turns through 150°; it is then to remain at rest while the cam rotates through the remaining 30°, when the same cycle of movement is to be repeated as long as desired. This is briefly expressed as follows:

Harmonic rise	120°
Rest	60°
Uniformly accelerated and retarded fall	150°
Rest	30°

Fig. 40. 1. *Follower Positions.* The length of travel Ae being known, it is divided for purposes of the rise, by the principles of Fig. 38, into distances giving harmonic motion of the follower; and for purposes of the fall, by the principles of Fig. 39, into distances giving uniformly accelerated and retarded motion. Eight spaces are chosen, so that the process of making these divisions is an exact repetition of the method in Figs. 38 and 39.

2. *Cam Radii.* The diameter of the base circle being chosen, and, for the purpose of this problem, the center being assumed in line with the path of the follower, draw the original radius C A, then the radii C X, C Z, and C W, limiting the arcs of rise, rest, and fall respectively. As eight divisions of the follower path have been chosen, the arcs of rise and rest must each be divided into 8 equal divisions. It should be observed that while the divisions are equal throughout each arc, the arc of fall being different from the arc of rise, the divisions of the arc of fall are *not* equal to the divisions of the arc of rise, measured on the base circle.

3. *Follower Rotation.* The rotation of the follower about the cam, for the period of *rise*, is accomplished by striking arcs through the points 1, 2, 3, 4, 5, 6, 7, and e, making the intersections R_1, R_2, R_3, R_4, R_5, R_6, R_7, and R_8, with the cam radii. The rotation of the follower about the cam, for the period of *fall*, is accomplished by striking arcs through points $1'$, $2'$, $3'$, $4'$, $5'$, $6'$, $7'$, and e, making the intersections F_1, F_2, F_3, F_4, F_5, F_6, F_7, and F_8 with the cam radii. These points of intersection represent the centers of the follower roll in its rotated positions, and from these centers should be struck arcs with a radius equal to the radius of the follower roll.

4. *Tangent Line.* A smooth curve is now drawn tangent to these small arcs, thus forming the outline of the cam.

5. *Testing.* The cam may be tested by the tracing-cloth method, precisely as in the cases already developed.

6. *Pressure Lines.* The pressure lines are found in exactly the same manner as in Fig. 31, and can be rotated back to the points in the path of the follower, in order to conveniently study the change in direction of the pressure lines as the follower moves along its path.

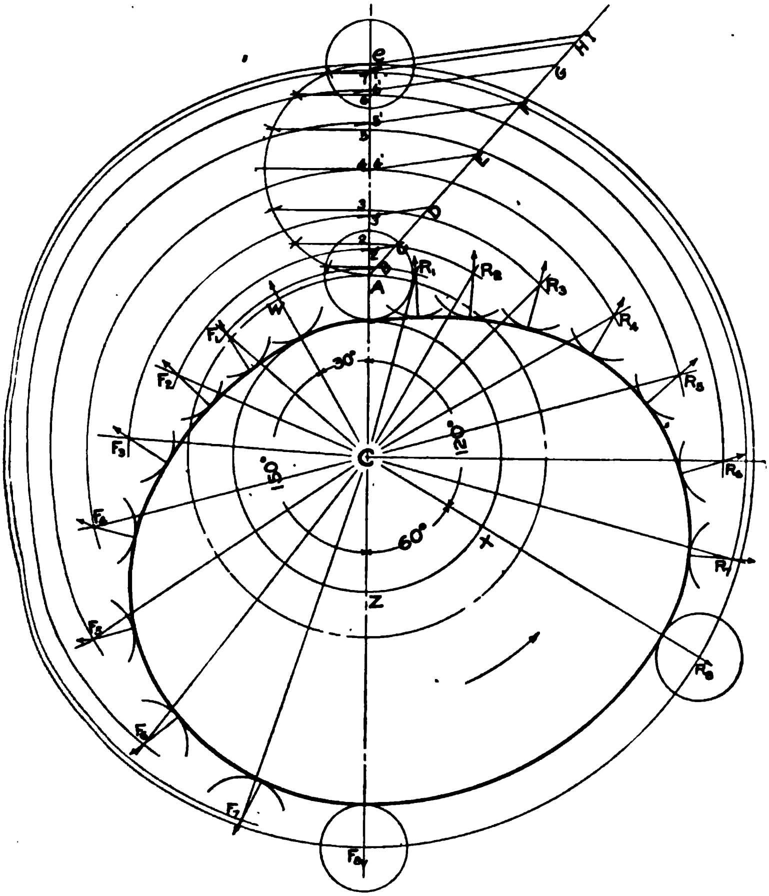


Fig. 40. Practical Problem with Rotating Plate Cam.

It should be noted that in Fig. 40 the same method of procedure is followed as in Figs. 29 to 36, except that the determined points in the path of the follower do not make *equal* divisions of the path, as in the case of uniform motion.

TRANSLATION CAMS

Rotating plate cams, like those thus far considered, are most commonly met with in practice. A straight-line, reciprocating motion of a plate, however, may be made to produce similar follower movements, in which case the cam is known as a *translation cam.* A straight-line movement is equivalent to movement along an arc with infinite radius. With this understanding, the same principles may be made to apply to translation cams as to rotating cams. Suppose it is required to produce the same movement of the follower as in Fig. 40, by means of moving a plate in a straight line instead of rotating it. This case is shown in Fig. 41.

Fig. 41. 1. *Follower Positions.* The same follower motion being required as in Fig. 40, the path is laid out exactly in the same way, the follower positions for the rise along path D 8-fulfilling the requirement of harmonic motion, and for the fall along path 1 F_8, fulfilling the requirements of uniformly accelerated and retarded motion. This is shown in the figure, and it is observed that no change from the method of Fig. 40 is employed.

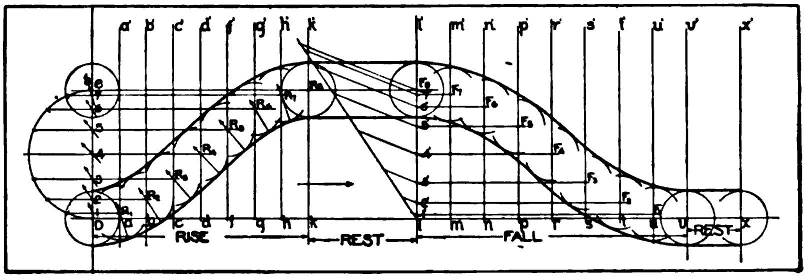
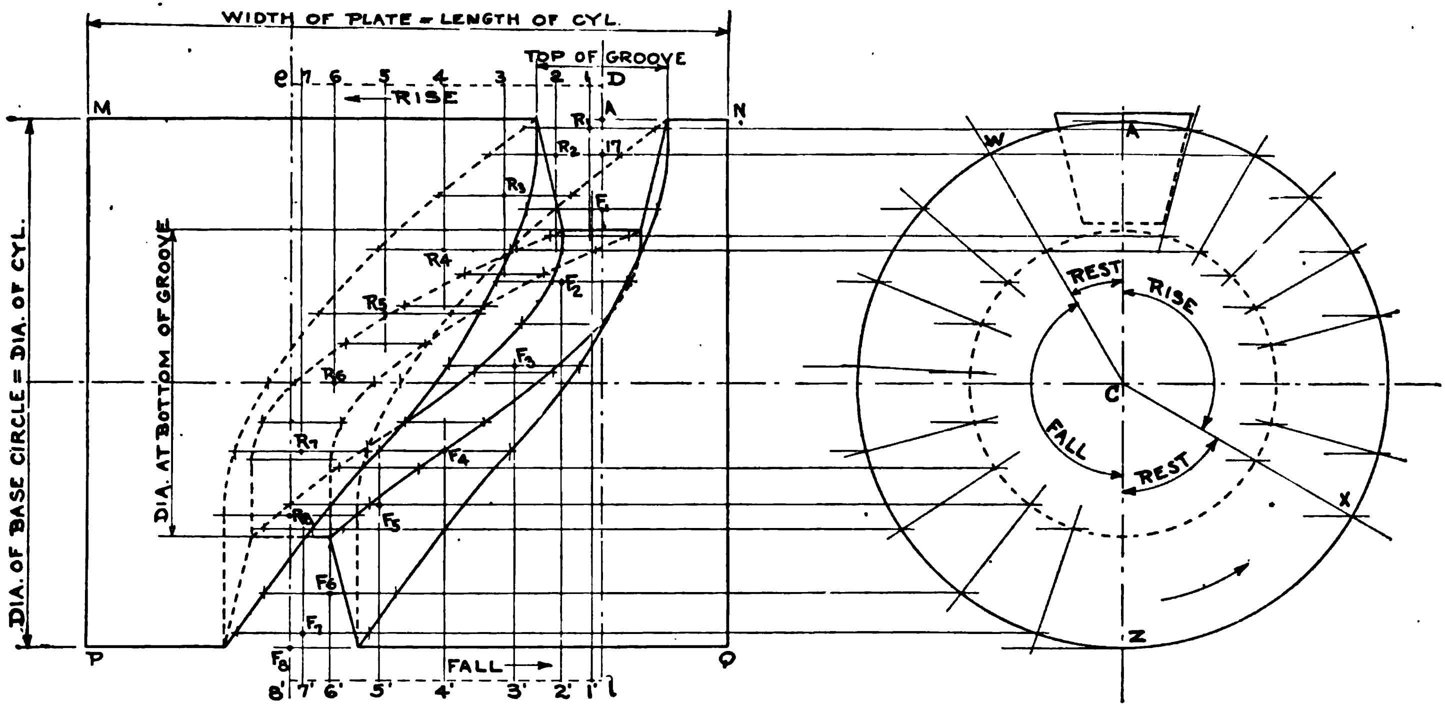


Fig. 41. Translation Cam.

2. *Cam Radii.* The base circle does not exist in this case as a circle, but has become a straight line, and may be chosen of any length, say D x. The cam radii, being always perpendicular to the cam arc (in this case the straight line D x), become parallel lines, perpendicular to D x. The cam arc of rise in Fig. 40 is now represented in Fig. 41 by the distance D 8, which should fulfill the relation $\frac{Dk}{Dx}=\frac{120}{360}$; in order to make the same relative movement of cam dur-



MARKS ON THIS DRAWING AGREE WITH THOSE ON FIG. 41.

Fig. 41a. Cylindrical Cam.

ing rise as in Fig. 40, D k should likewise be divided into 8 equal parts. The arc of rest in Fig. 40, being ½ the arc of rise, the distance k l in Fig. 41 is made ½ the distance D k. The arc of fall in Fig. 40 being 1¼ the arc of rise, the distance l v in Fig. 41 is made 1¼ the distance D k. The final arc of rest in Fig. 40 being ½ the first arc of rest, the distance v x in Fig. 41 is made ½ the distance k l. This completes the cycle; and the parallel lines aa′, bb′, cc′, etc., drawn through the several points of division as noted, represent the several positions of the cam radii.

3. *Follower Rotation.* Since the lines of follower rotation are all perpendicular to the cam radii—which in this case, are all parallel—the rotation, or translation, of the follower is accomplished by drawing parallel lines through the determined points of the path, producing the intersections R_1, R_2, R_3, R_4, etc. Between points R_8 and F_8, the follower rests; and for the period of fall, the intersections F_8, F_7, F_6, F_5, etc., are determined as for the rise, by producing the parallel lines through the points in the path of fall. From point v to x the follower again rests. These intersections represent the centers of the follower in its *translated* positions.

Now, with radius equal to the radius of the follower roll, arcs are struck to represent the outline of the follower in each of its translated positions.

4. *Tangent Line.* A smooth curve is now drawn tangent to the several translated positions of the follower roll. In this cam a new feature is introduced by drawing these tangent lines on both sides of the roll, thus making a groove which holds the follower firmly in position at all times. This gives an absolutely positive fall to the follower roll. The same grooved construction might have been made on any of the cams heretofore studied, instead of allowing the follower to come down by gravity or the force of a spring.

5. *Testing.* The cam may be tested by the tracing-cloth method as before, the procedure in this case, however, being one of translation instead of rotation. The original radius with the follower in its several positions being traced upon the cloth, is set upon each of its translated positions, and, by careful inspection, it is noted whether the roll, in this position, just touches the faces of the cam groove as drawn.

6. *Pressure Line.* The pressure lines are drawn precisely as in all cases thus far considered, and may be translated back to the path of the follower in order to study their direction as the follower moves along its path.

Although the same cycle of follower movement has been accomplished in this case as in the rotating cam, Fig. 40, the translation cam is not in position to begin a repetition of the cycle by further movement. If we reversed the motion of the cam the cycle also would be reversed; and in the cam under discussion we should have a rest, then a uniformly accelerated and retarded rise, then a rest, then a harmonic fall, the periods of time being reversed as well as the motion. We could, of course, by choosing the motion for rise and fall exactly the same, secure the same motion for the reversed as for the forward movement of the cam.

CYLINDRICAL CAMS

Suppose that the outline of the translation cam as developed in Fig. 41 be wrapped around a cylinder whose circumference is exactly equal to D x, and that the lines represent a spiral groove cut into the surface of the cylinder. If, then, a follower roll be allowed to remain in this groove while the cylinder is rotated on its axis, the cycle of follower movement will be repeated as long as we choose to rotate the cylinder. Such a grooved cylinder is known as a *cylindrical cam.*

Fig. 41*a* shows a cylindrical cam in two projections. Attention is called to the tapering follower roll used. This is because it is necessary that points on the sides of the groove and the surface of the roll have the same velocity about the center C. The roll, therefore, must be the frustum of a cone whose apex is at C.

The top of this groove is produced by wrapping Fig. 41 around the surface of the cylinder as previously described. The bottom of the groove is produced by making a translation cam whose base line is equal in length to the circumference of the cylinder at the bottom of the groove, and then wrapping it around that cylinder. This base line for this smaller cylinder is divided into the same number of parts as the line D x, the divisions, however, all being proportionally smaller. The length of the follower path, and its divisions, are in nowise different from those shown in Fig. 41; and the method of developing the outline of the cam is precisely the same.

The wrapping of these translation cams, for purposes of the drawing, is accomplished by means of dividers and compasses, according to the principles for the development of cylinders, as explained in Part III.

The limitations of construction of cylindrical cams are considerably greater than those of the simpler rotating cams; and it is more frequently a question of experiment and trial to get the proper surfaces, than it is of exact theoretical lay-out on the drawing-board

BELTING

Cams and gears transmit positive motion from the driver to the follower by direct contact of the surfaces. As the distance between centers of shafts increases, the driver and follower for such methods of transmission become large, unwieldy, and costly, and rigid links may be used to connect the rotating pieces, as in the case of parallel rods of a locomotive. For a further increase of distance, the transmission is attained by means of belts and pulleys, and, if the distance is very great, by wire ropes and sheaves. As there is always some slipping of the belt (from 1 to 2 per cent), the velocity ratio is not exact; but this is not essential in many classes of machinery. The

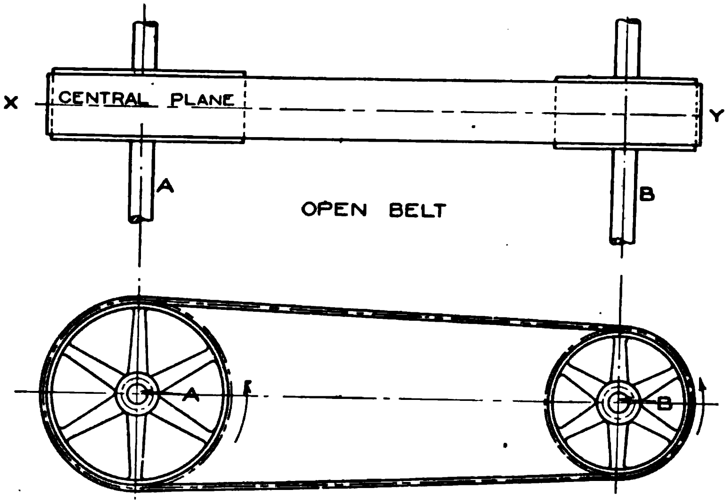


Fig. 42. Belt Drive with Open Belt.

slip and stretch of the belt reduce the shock when heavy machinery is set in motion—an important feature in many cases.

Open and Crossed Belts. The simplest forms of belt-drives are the *open belt* (Fig. 42) and the *crossed belt* (Fig. 43). In each case the shafts are parallel, and the pulleys fastened to the shaft with set screws or keys. The central planes of the pulleys must obviously be coincident. The belt is then tightly stretched over the pulleys, and, assuming B, the driver, to turn in the direction of the arrow, motion will be transmitted to A, on account of the friction set up between the belt and pulley surfaces. The fibers of the belt, in running on or off the pulley, bend over one another, so that those next the pulley, on the inside of the belt, are compressed, while those on the outside are stretched. Assuming the compression and stretch to be equal, then the central fiber does not change in length. This central fiber is shown in the figure by a "dash-and-dot" line. Considering that there is no slip of the belt on the pulley, the face of each pulley will move exactly with the belt, and the turns of each pulley will depend on its circumference; or,

$$\text{Turns of A} = \frac{\text{Speed of belt}}{\text{Circumference of A}} = \frac{S}{\pi \times \text{Diameter of A}};$$

$$\text{Turns of B} = \frac{\text{Speed of belt}}{\text{Circumference of B}} = \frac{S}{\pi \times \text{Diameter of B}};$$

$$\text{Velocity ratio} = \frac{\text{Turns of A}}{\text{Turns of B}} = \frac{\text{Diameter of B}}{\text{Diameter of A}}.$$

Thus the velocity of the shafts is inversely proportional to the ratio of the diameters of the pulleys. The action of the belt in bending about its central fiber has the effect of increasing the diameter of the pulley by an amount equal to the thickness of the belt, and an *exact* calculation for velocity ratio must take this fact into consideration. For example, suppose that the diameters of A and B are 8 inches and 24 inches respectively, and that the belt is ¼ inch thick. Then the velocity ratio is $\frac{24}{8}=3$ for the *usual approximate calculation*; but $\frac{24.25}{8.25}=2.939$ for the exact value.

The direction of shaft rotation depends on the method of applying the belt. In the case of the open belt, the top surfaces of each pulley being connected, each shaft rotates in the same direction; while in the case of the crossed belt, the top surface of A being connected to the bottom surface of B, the shafts rotate in opposite directions. Thus the directions of rotation are the same when the center-line of belt lies wholly on one side of the line connecting the centers of pulleys; and different when the center-line of belt intersects the line of centers.

Crowning Pulleys. Suppose that a flat belt is placed on the side of a double cone, Fig. 44, and that we start to rotate the cone in the direction of the arrow. The edge E, which is stretched more tightly than F, has a greater grip on the surface of the cone, and will climb up the incline as shown by the dotted lines. With continued rotation of the cone, the belt, if not prevented, will move further up the incline, will finally pass the crest and start down on the other side until the two edges E and F have equal tension, or the pull to the right is balanced by an equal pull to the left. The center line of the belt will now run in the central plane X Y of the pulley.

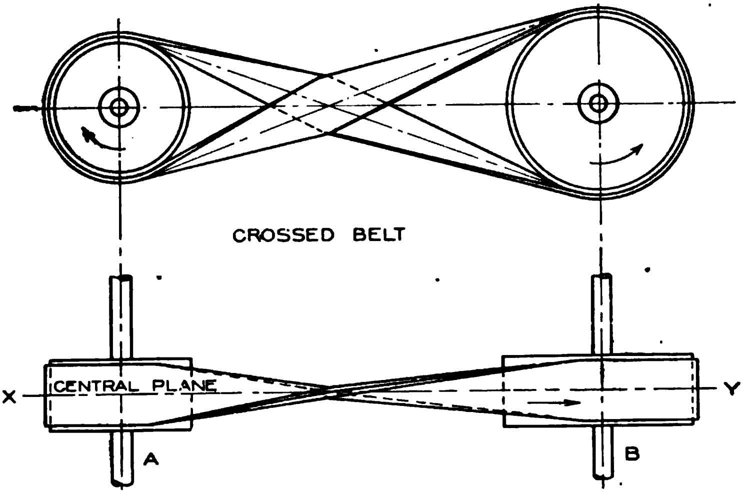
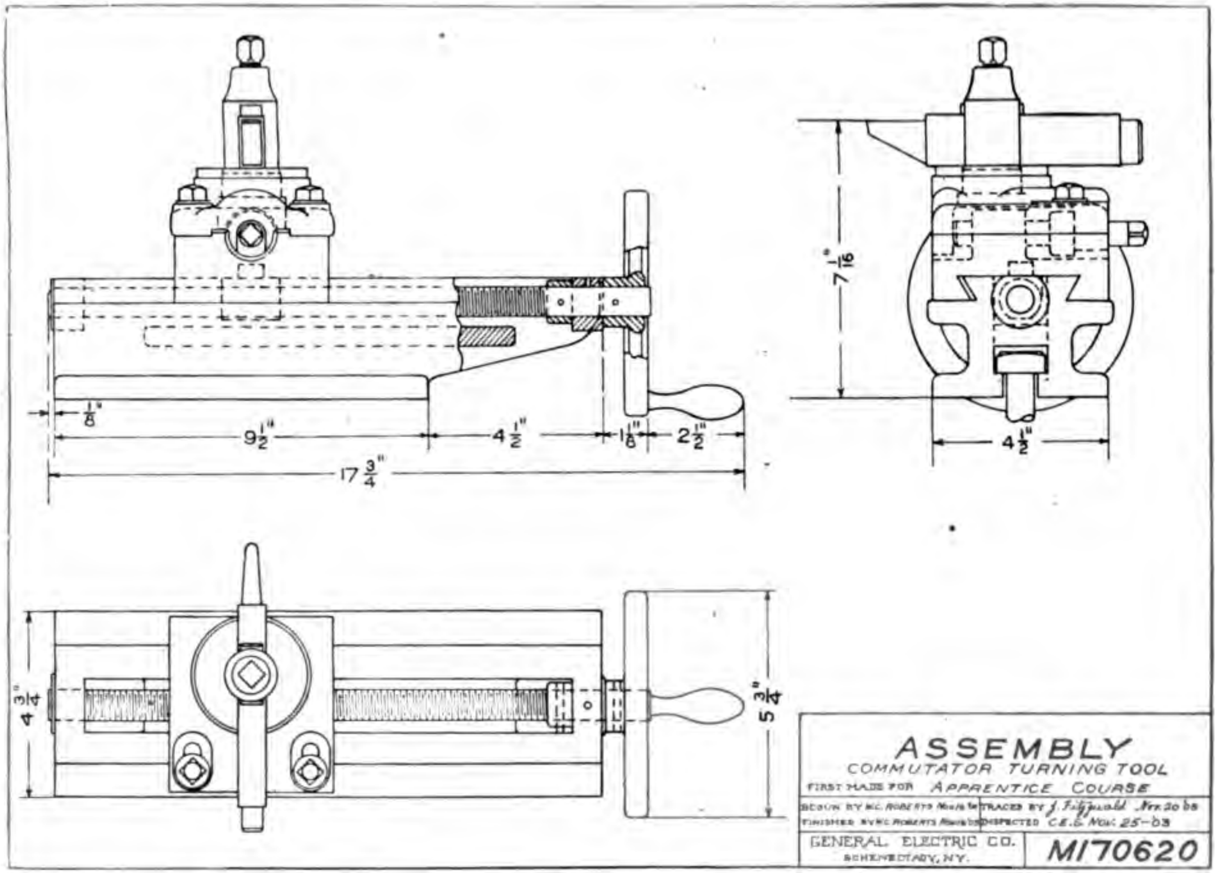


Fig. 43. Belt Drive with Crossed Belt.

As long as this condition is maintained, the belt will run true and will stay on the pulley. Also, if the pulley faces were perfectly flat, the belt a perfectly homogeneous piece of leather, and the shafts perfectly parallel, the belt would stay in the pulley. Such perfect conditions, however, cannot be produced or maintained, and it is therefore necessary in practice to imitate the conditions of Fig. 44, and *crown* either one or both pulleys, so that the belt will not develop any tendency to run off. On the contrary, it will constantly seek


1/8"
9 1/2"
4 1/2"
1 1/8"
2 1/2"
17 3/4"
7 1/16"
4 1/2"
4 3/4"
5 3/4"
ASSEMBLY
COMMUTATOR TURNING TOOL
FIRST MADE FOR APPRENTICE COURSE
DESIGN BY W.C. ROBERTS Nov.18 '08
TRACED BY J. Fitzgerald Nov. 20 '08
FINISHED BY W.C. ROBERTS Nov.18 '08
INSPECTED C.E.G. Nov. 25-08
GENERAL ELECTRIC CO.
SCHENECTADY, N.Y.
MI70620

EXAMPLE OF SHOP DRAWING MADE IN DRAFTING ROOM OF THE GENERAL ELECTRIC COMPANY.

to keep its center line in the central plane of the pulleys, any tendency of one edge to slacken and run off being instantly counteracted by the tightening up of the other as it starts to climb, thus pulling the belt back until the balance is secured. A very slight amount of *crowning* will accomplish this result, and as little crown as possible, consistent with good running, should be provided, that too great inequality of tension in the belt may not be introduced.

In Fig. 42, if the shafting be not parallel, the center line of the belt will not run in the central plane of the pulley, and the belt will climb towards the high part of the pulley, as in the above case of the cone, and we may expect the belt to run off.

The above discussion reduces to one fundamental working condition for belt transmission; viz., *The center line of the belt leaving a pulley must lie in the central plane of the pulley to which the belt is delivered.*

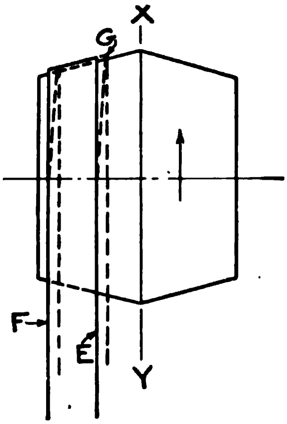


Fig. 44. Crowning Pulley.

This principle is applicable to all cases of belt transmission, however complicated, whether the shafts be parallel or at an angle. The use of guide pulleys is merely a means of controlling the delivery of the belt according to the principle. The student should commit this principle to memory, and go over its application until he is sure that he understands it; for without it he cannot solve complicated belt problems, and he cannot feel sure of his solution of even the simplest ones. In the pages of discussion and illustration which follow, the student should constantly note the recurrence and application of this principle, as it is really all there is to the solution of belt problems, except familiar knowledge of how to make the drawing projections, which he is already supposed to possess.

Tight and Loose Pulleys. Tight and loose pulleys are provided for cases in which a machine is to be thrown in and out of service without stopping the driving shaft.

A common arrangement is shown in Fig. 45, where a pulley with a straight face is located on the line shaft, and two pulleys with crowned faces are on the counter-shaft. Pulley B is loose on the shaft, and C is fast to the shaft. A collar D is placed on left of B, to prevent its end motion. Shifting the belt is accomplished by pushing on the *advancing side of belt,* close to the receiving pulley. The shifter F has two iron pegs, one on each side of the belt, and is operated by hand. The belt is readily guided by pressure on the advancing side, for the motion of the receiving pulley aids the motion of the belt; but if pressure were applied to the retreating side it would be necessary to use enough force to shift the belt bodily

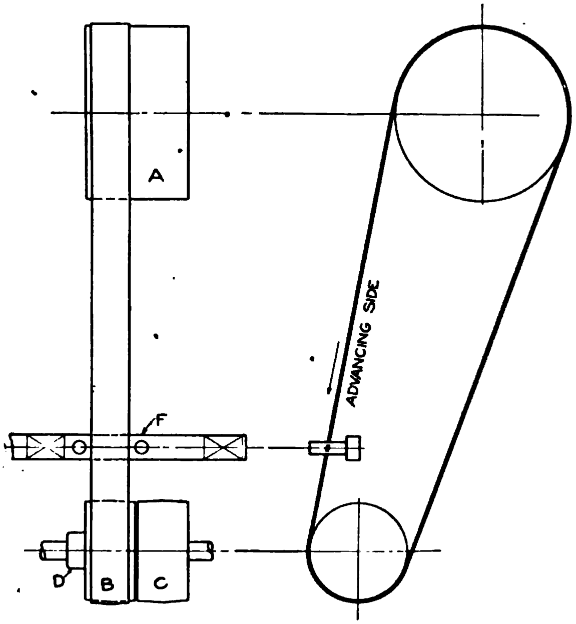


Fig. 45. Straight-Faced Pulley on Line-Shaft; Crowning Pulley on Counter-Shaft.

sidewise on the face of the pulley. As excessive tension on the belt decreases its life, the diameter of the loose pulley is often slightly decreased, thus permitting the belt when doing no useful work to be under less tension than when driving.

Another arrangement is to provide a clutch to throw into a loose pulley on the line shaft, the belt standing idle when not in service.

Shafts not Parallel. Suppose an open belt to connect pulleys A and B_1 on parallel shafts, Fig. 46.

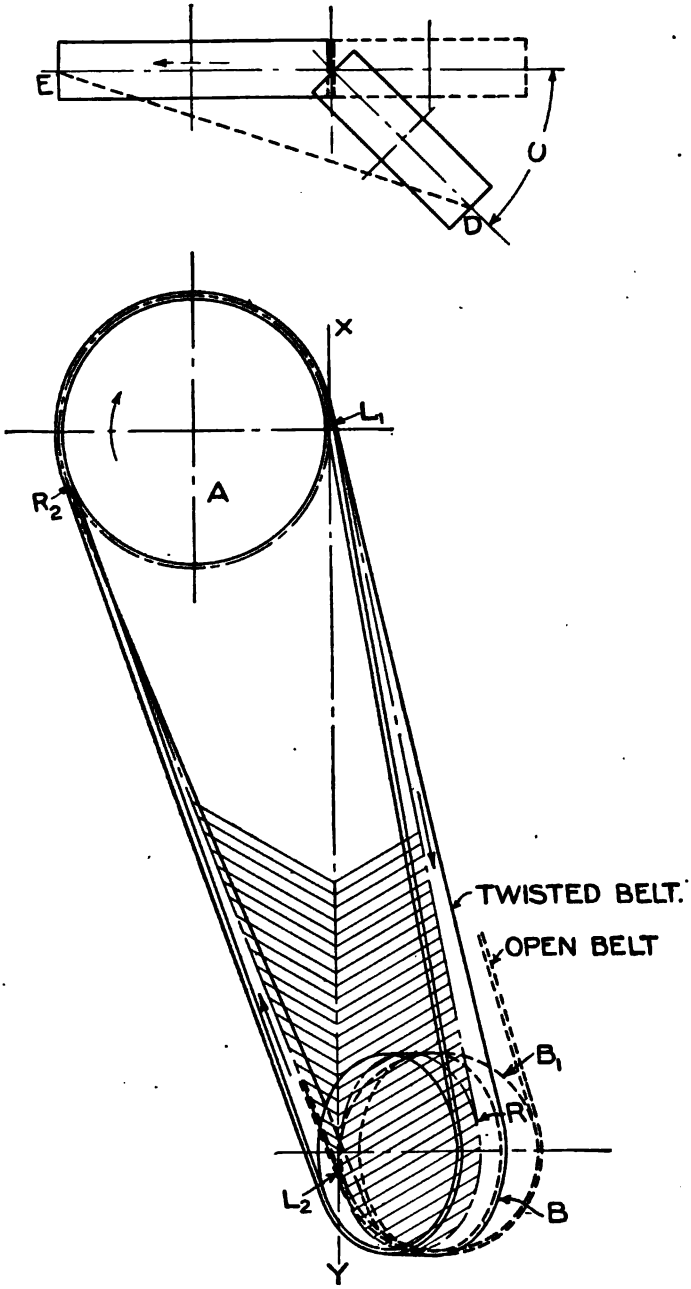


Fig. 46. Open Belt Connecting Pulleys on Shafts not Parallel

Draw a tangent XY to the pitch circles of the pulleys at the points L_1 and L_2, where the belt *leaves* the pulleys. Now rotate the central plane of the pulley B_1 about XY as an axis, through any angle C, to position shown by pulley B. The central planes, (shaded) of pulleys A and B intersect on the line XY, called the *trace* of the planes. The axes are now not parallel, but the belt may be made to run in one direction, for it still obeys the general principle of the guiding of belts; *i.e.*, the center line of the belt, on leaving the driving pulley, is delivered into the central plane of the receiving pulley.

Examining the figure, we find that the center line of the belt moves in direction of arrow from L_1 to R, and around pulley B to L_2, from L_2 on the surface of B to R_2, thence on surface of A to L_1, the starting point. From the point L_1, where the belt leaves A, until it reaches R, the center line of the belt is in the central plane of the receiving pulley B, and the belt twists about this line, presenting a flat side to the face of pulley B at R.

From L_2, where the belt leaves B, until it reaches R_2, the center line of the belt is continually in the central plane of the receiving pulley A, and a similar twist in the belt takes place. If now we attempt to reverse the direction of motion of the belt, the top of pulley A, moving in the direction of the dotted arrow, would carry point D of the center line of belt to the left-hand edge of A, as indicated by the dotted line D E, where it would drop off. Therefore this belt-drive for shafts not parallel is suitable only for motion in one direction.

Quarter-Twist Belt. By rotating the central plane of B, Fig. 46, until the angle C becomes 90°, a *quarter-twist*, or *half-crossed* belt, Fig. 47, is obtained; and if angle C becomes 180°, the crossed belt of Fig. 43 is the result.

If the thickness of belt is neglected, it will be noted that the central plane of B (Fig. 47), which is represented by the trace X Y, is tangent to the surface of pulley A; and similarly on the side view, the trace S T of the central plane of A is tangent to the surface of B. Now follow the center line of belt in the direction of the arrows on the front and side views, starting at point A, where the belt leaves the upper pulley. From A to D, and around to the back of pulley B, the center line is shown in the trace X Y of the front view. From

B, where the belt leaves the lower pulley, to C, and around to the front and starting-point A, the center line is shown in the trace S T of side view. Therefore the center line of the belt is always delivered into the central plane of the receiving pulley, and it will drive satisfactorily in the direction of the arrows. Reversing the motion will cause the belt to run off the pulleys, as in the previous case.

The belt should always be put on so that the same side of the belt touches both pulleys when it is possible to do so. In making the drawing of the belt we shall call the side of the belt which touches the pulleys the *inside*, and the other side the *outside*. Now, referring first to the left-hand elevation in Fig.47, where the belt lies around the circumference of the upper pulley, from the point C, where the upward-moving part of the belt strikes the pulley, around to A, where the downward-

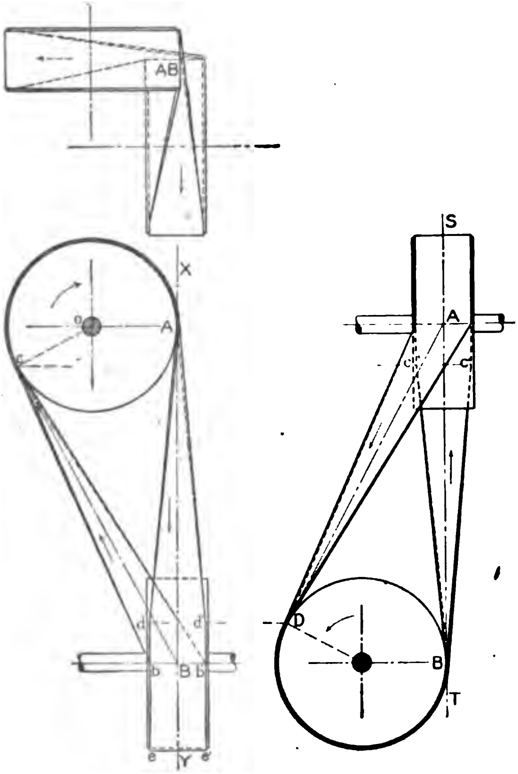


Fig. 47. Quarter-Twist Belt.

moving part leaves the pulley, only the edge of the belt is visible, and is represented by drawing an arc of a circle from line O C around to O A, with a radius equal to the radius of the pulley plus the thickness of the belt. The location of C may be found closely enough for all practical purposes, by drawing a line from B tangent to the upper pulley, C being the point of tangency, and O C the radius drawn through C. In drawing the arc of the circle from O C to O A, it is well to let it run by these lines a little way in the pencil drawing, and that part which is not used may be erased after the drawing is inked. The descending part of the belt leaves the pulley at A;.and from that point to the place where it strikes the lower pulley, it twists through an angle of 90°, coming out over the front of the lower pulley. The edge of the belt, which we see in its full thickness where it leaves the upper pulley at A, twists toward the left, less and less of it being seen as the belt descends, until at *d*, where it strikes the other pulley, the two corners of this edge coincide, and from there to the bottom of the pulley we see this edge of the belt as a line. At the same time that the thickness of the belt has been disappearing from view, the outside of the belt has been coming into view, until at *d d'* we see the full width of the belt, the outside corner being the one which is visible. The inside corner is behind the rest of the belt, and is shown dotted in the figure, although it is usually omitted entirely. From *d'* to the bottom of the pulley the two right-hand corners coincide. Points *d* and *d'* are found by drawing the horizontal dotted line through D (side view), and laying off on this line a distance equal to one-half the width of the belt each side of line X Y. From *e e'* the belt goes around the back side of the pulley to *b b'*, where it begins to be drawn off to the left, *b b'* being on the center line drawn through the shaft, since, by glancing at the right-hand view, we can see that the upward-moving part of the belt leaves the lower pulley at B, which is on a level with the center of the shaft. After leaving *b b'*, the belt remains behind the pulley for a short distance, and is shown dotted. It is projected at its full width at *b b'*, but as soon as it leaves the pulley it begins to twist toward the right, the left-hand edge gradually coming into view until it is seen at its full thickness where it strikes the pulley at C. At the same time, the *inside* face of the belt, which is toward the front when the belt leaves the pulley, gradually turns toward the right, and appears narrower until it disappears from view—that is, is projected as a line. This completes the front view, and the side view is drawn according to the same principles, as is also the plan view.

To be strictly correct, the sides of the belt should be shown curved at A and B, for the belt is pliable and starts to curl and slip sidewise a short distance above these points. In order not to have an excessive side slip, the angles C B A and D A B should not be more than 25 degrees.

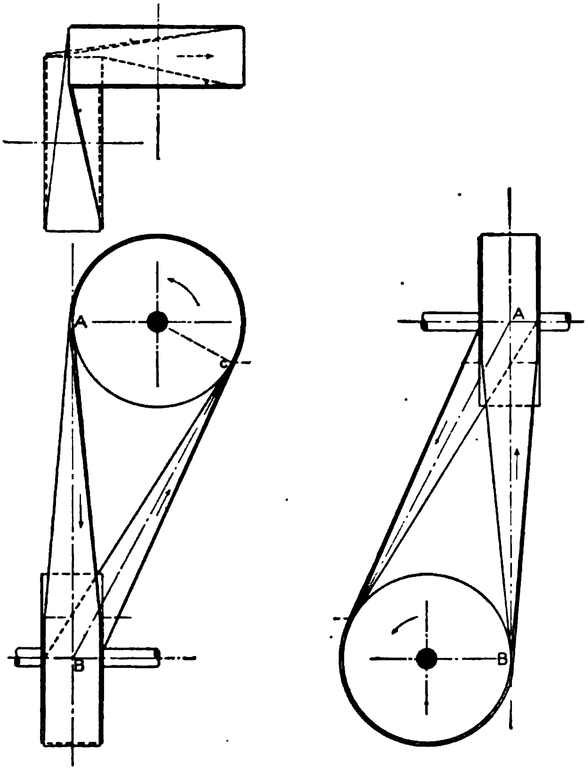


Fig. 48. Quarter-Twist Belt.

Figs. 48 and 49 show what changes are made in the location of the pulleys and the appearance of the belt by changing the direction of rotation of the shafts. In Fig. 48 the lower shaft turns in the same direction as in Fig. 47, but the upper shaft turns in the opposite direction. In Fig. 49 the upper shaft turns in the same direction as in Fig. 47, and the lower shaft turns in the opposite direction.

Reversible Quarter-Twist with Two Guide-Pulleys. In order to reverse a quarter-twist belt, it will be necessary to introduce one or two guide-pulleys to bring the center line of the belt at all times into the central plane of the receiving pulley. Fig. 50 shows an arrangement where two guide-pulleys are used. Let the driving pulley R and the driven pulley P be located as in the case of the quarter-twist belt of Fig. 48. Suppose that the direction of rotation is but seldom reversed, and that the usual motion of the driver is in the direction of the arrow. The part of the belt which has the greater load should have a direct connection between R and P, leaving the slack side to run over the guide pulleys, thereby decreas-

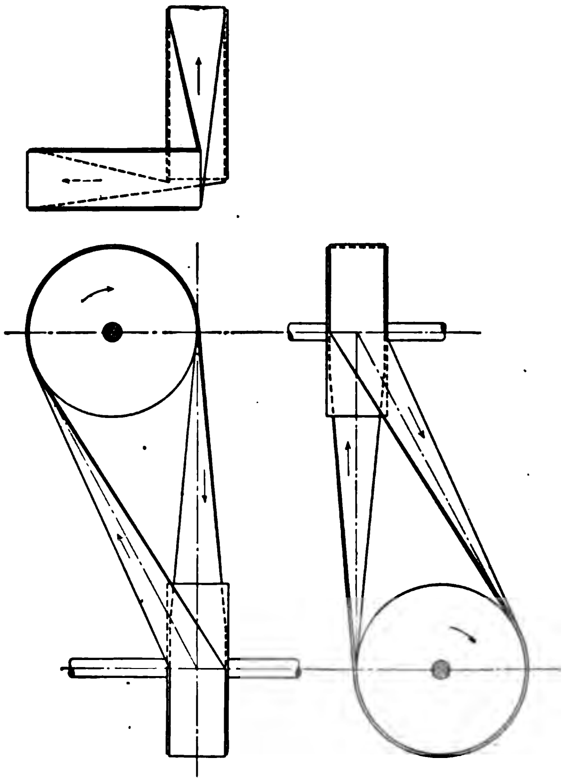


Fig. 49. Quarter-Twist Belt.

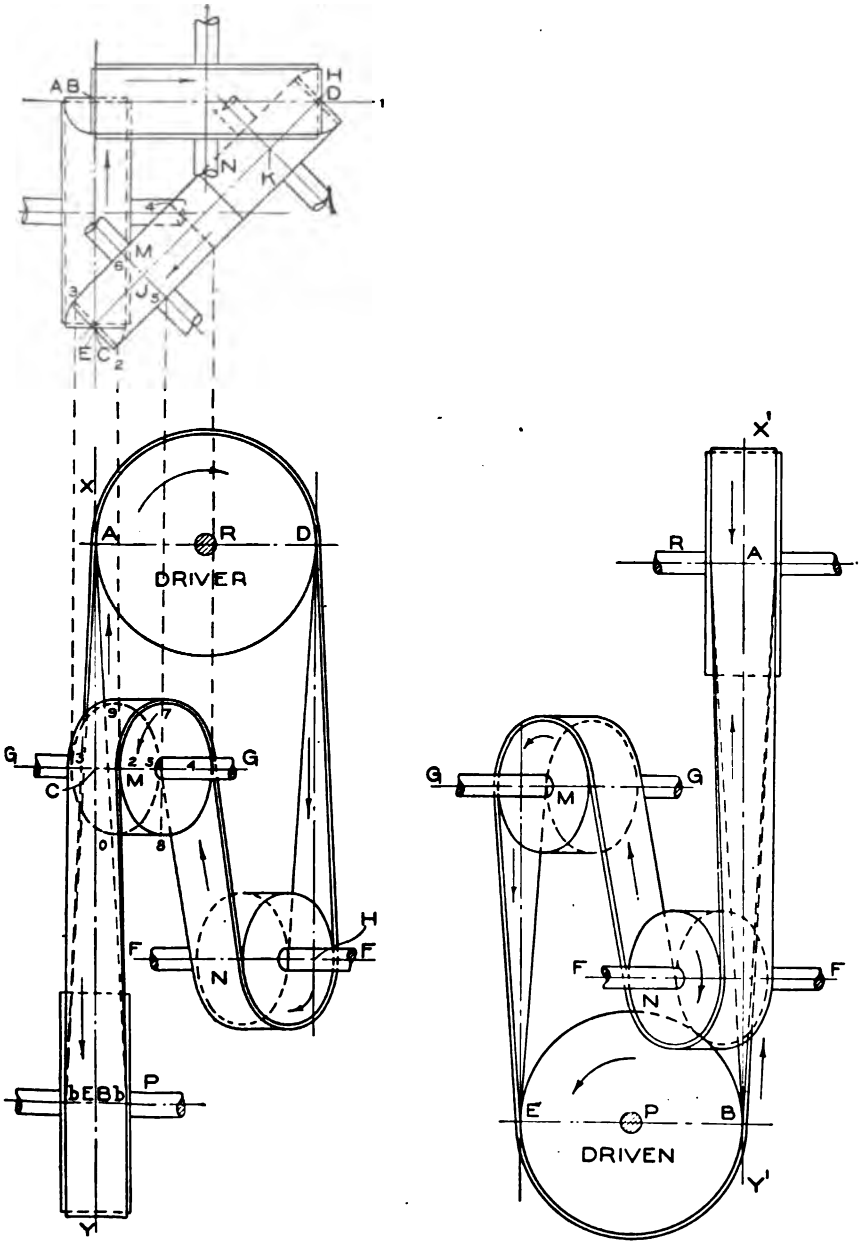


Fig. 50. Reversible Quarter-Twist Belt with Two Guide-Pulleys.

ing the stress on the belt and lessening the friction in the journals of the guide-pulleys.

Receiving point A is then connected to B, the point of departure from the back of the driven pulley, instead of over to the front face, as in case of Fig. 48; and the belt twists about this line as an axis through an angle of 90°, so as to present flat sides to both pulleys. The plan view of the center line of the belt is the line of intersection A B of the traces of the central planes of R and P. Evidently the belt will run in either direction along the trace of either plane. From the point of departure D of the driving pulley, the belt must be led to the receiving point E of the driven pulley, by means of guide-pulleys. Draw D E for the plan view of the central plane of the guide-pulleys. This plane is perpendicular to the paper, and its traces or intersections with the central planes of R and P are shown in X Y and D H of the front view.

The guide-pulleys are idlers introduced only to form a path for the belt, and do not in any way affect the velocity ratio. They may, therefore, be made of any convenient size to suit the existing conditions. We may then assume on the plan view, that J and K are points on the axes of the pulleys M and N, and draw the guide-pulleys so that the face of M bisects the front face of P at E, and one face of N bisects the right-hand face of R at D. On the front and side views, the location of the axes has been assumed as G G and F F, and the corresponding views of the pulleys drawn according to the principles of projection.

Examining the belt in passing from D to E, we find its center line goes from D in the central plane of R, to H in central plane of N, around N and M, which have one common central plane, arriving at C, a point in the trace of the central planes of M and P, and from *leaving* point C to *receiving* point E. It will be noted that the center line of belt connecting both leaving and receiving points is always in the central planes of both pulleys, and therefore the direction of motion may be reversed at will. If the belt be followed around the pulleys, we find both sides of the belt come, successively, in contact with the pulleys. Therefore, the belt must be given a single twist before uniting the ends together.

It is usually desirable to have only one side the working side; but if such were the case in this arrangement of pulleys, it would

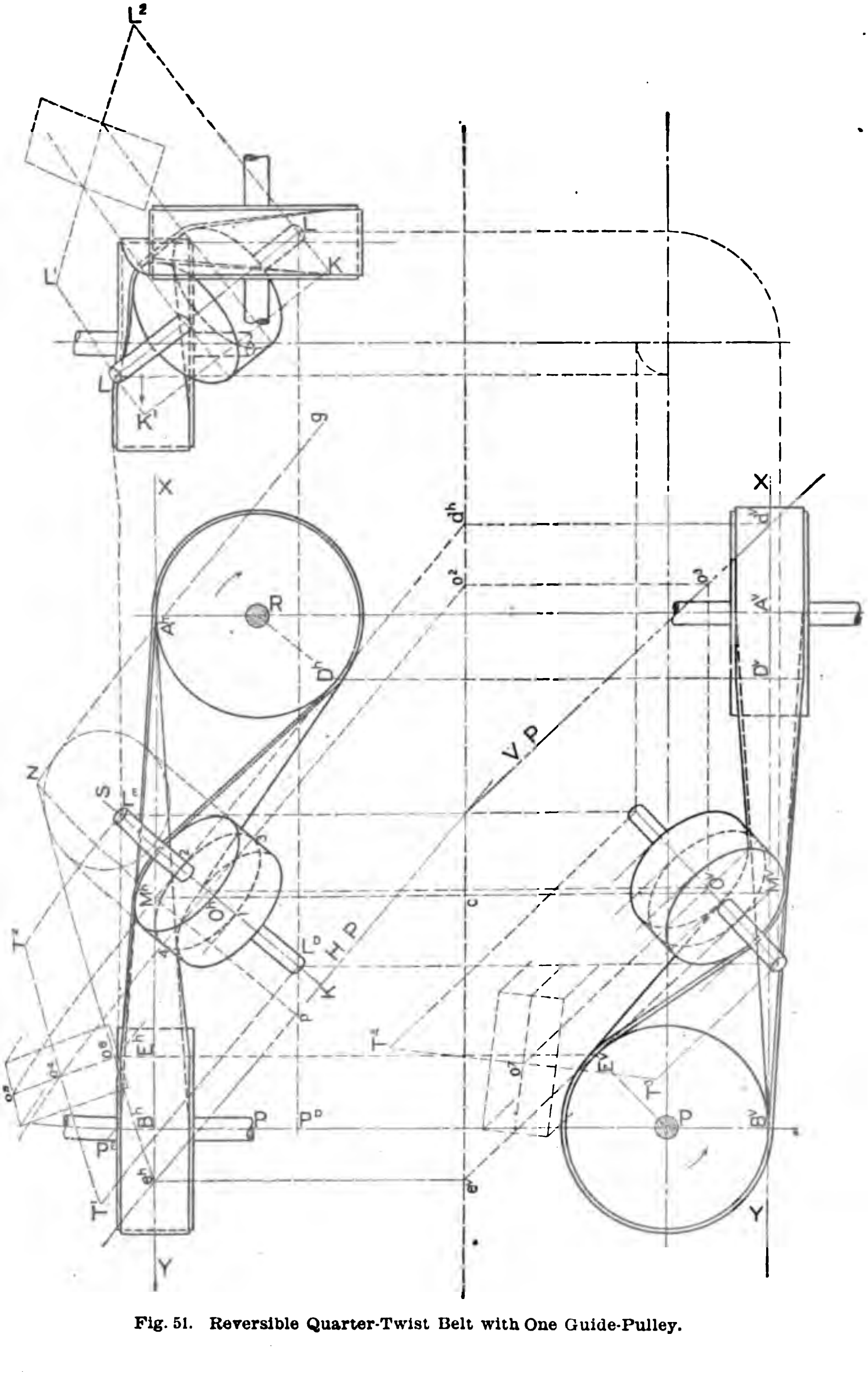


Fig. 51. Reversible Quarter-Twist Belt with One Guide-Pulley.

bring a sharp twist in the belt, between M and N, which might be a greater objection.

Reversible Quarter-Twist with One Guide-Pulley.* Fig. 51 shows the arrangement of the pulleys for the belt to run in either direction, using only one guide-pulley. The ordinary direction of rotation is that shown by the arrows. The upper pulley is the driver, and, as in the case where two guide-pulleys are used, the tight part of the belt goes directly from one main pulley to the other, the slack part of the belt returning over the guide-pulley. The shaft of the guide-pulley must be set at an angle with both main shafts in order to guide the belt properly. The method of locating the main pulleys is exactly the same as in the case where the two guide-pulleys are used, so that the explanation given for that case will apply here, and all we need to consider in this case is the location and drawing of the guide-pulley.

We shall first consider the two elevations. The plumb line X Y, as well as being the center line of the tight part of the belt, is the line of intersection of the central planes of the two main pulleys. Choose a point in the line X Y, which may be anywhere along the line, depending on how far the guide-pulley is to be from one or the other of the main shafts, but preferably about half-way between them. The point is marked M^h in the left-hand elevation, and M^v in the right-hand elevation. From M^h draw a line tangent to the upper pulley at D^h, and from M^v draw a tangent to the lower pulley at E^v. The other projection of the line M^h D^h will be M^v D^v, coinciding with X Y in the right-hand elevation; and the other projection of M^v E^v will be M^h E^h, coinciding with X Y in the left-hand elevation. We now have two lines, M D and M E (shown respectively by their two projections, M^v D^v—M^h D^h and M^v E^v—M^h E^h), which determine the plane of the guide-pulley and which are practically the center lines of that part of the belt which passes over the guide-pulley, and our problem is one of projections. The problem is to find on the drawing paper the traces of the plane which contains the two lines, which is the central plane of the guide-pulley, and revolve this plane about one of its traces until it is parallel to the plane of the paper, so that the true angle between the lines will be shown; then draw the guide-pulley tangent to the lines in their revolved position, and revolve the lines back to their former position, revolving the guide-pulley back at the same time. To carry out this construction, proceed as follows: Draw a ground line anywhere between the two elevations, parallel to X Y; and for the time being consider one of the elevations as a horizontal projection and the other as a vertical projection, remembering that our drawing is made as if projected on two planes located as in Fig. 2, Part IV, or, as it is commonly expressed, "in the third quadrant." We shall treat the left-hand elevation as if it were the horizontal projection, and the right-hand elevation as if it were the vertical projection. Extend line M^h D^h until it meets the ground line at d^h; and at d^h draw a perpendicular to the ground line, meeting X Y (which is the same as M^v D^v extended) at d^v. Through d^v draw a line parallel to M^v E^v; and this line, which is marked V P, is the vertical trace of the plane which contains the lines M D and M E. In like manner find the horizontal trace by extending M^v E^v to meet the ground line at e^v, erecting a perpendicular at e^v to meet M^h E^h at e^h, and drawing H P through e^h parallel to M^h D^h. If the work is correctly done, H P, V P, and the ground line will intersect in a common point. Now through M^h draw a line perpendicular to H P, meeting H P at *p*. Construct the right triangle $t\,m^h$ P (Fig. 52), making m^h P equal to M^h p on Fig. 51, and making $m^h\,t$ equal to the perpendicular distance of M^v from the ground line in Fig. 51 (that is, equal to $c\,M^v$). Then take the distance P *t* (Fig. 52) and lay it off on the line p M^h (Fig. 51) from *p*, thus obtaining point N. Join N and e^h, and through N draw N *g* parallel to M^h D^h. The lines N e^h and N *g* are the projections on the horizontal plane of lines M E and M D, respectively, when the plane P, which contains these two lines, is revolved so that it is parallel to the horizontal plane. Therefore the angle *g* N e^h is the true size of the angle between lines M E and M D. Now, with a radius equal to the radius of the guide-pulley which is to be used, draw a circle which shall be tangent to the lines N e^h and N *g*. This circle is the central circle of the guide-

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*This subject is optional.

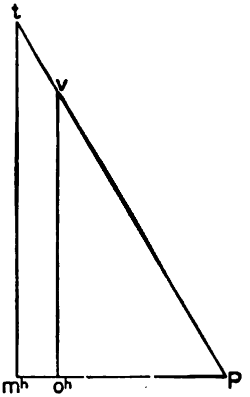

Fig. 52.

pulley revolved parallel to the horizontal plane, and its center S is the revolved position of the center point of the guide-pulley, and of course lies in the plane P. To revolve the central circle of the guide-pulley back so as to get its two projections when it is in the position which it actually occupies with relation to the two main pulleys, we shall first revolve the point S back to O^h. To do this, draw S K perpendicular to H P. Then in Fig. 52, lay off from P along the line P *t* the distance P V, equal to S K in Fig. 51. Care must be taken to lay off this distance from P rather than from *t*; and in order to remember from which point to measure, the student can bear in mind that distances measured along the hypothenuse from P (Fig. 52) represent distances measured from H P (Fig. 51). Having thus found point V, draw a line perpendicular to m^h P, meeting it in o^h; take distance o^h P in the dividers, and lay it off from K along K S (Fig. 51), thus getting O^h. Then point O^h will be the center of the ellipse which represents the center circle of the guide-pulley in its actual position. Point 2, where K S cuts M^hD^h, will be one end of the minor axis, and point 1, found by laying off O^h 1 equal to O^h 2, will be the other end of the minor axis. The major axis is found by drawing a line through O^h parallel to M^hD^h, and laying off along this line from O^h the distances O^h 3 and O^h 4, each equal to the radius of the guide-pulley. Having now found the two axes of the center ellipse, it can be drawn by any geometric method for constructing an ellipse.

We shall next find O^v by prolonging the major axis of the ellipse just found until it meets the ground line at o^2, then erecting a perpendicular to the ground line at o^2 to meet V P at o^3, drawing a line through o^3, parallel to the ground line, and from O^h drawing a line perpendicular to the ground line which will meet the parallel through o^3 at O^v. This point will be the vertical projection of the center of the middle circle of the guide-pulley. The ellipse, which is the vertical projection of this middle circle, is found from O^v in a way exactly similar to that in which the ellipse for the horizontal projection was found from O^h.

The next step is to draw the guide-pulley and its shaft, and to do this we shall revolve the central ellipse over in each view in such a way that we shall have it projected as a line. We shall take the horizontal projection first. Extend the major axis of the ellipse from O^h to o^4, making O^h o^4 equal to the perpendicular distance of O^v from the ground line. Draw O^6 1 through point 1 parallel to O^h o^4. With o^4 as a center, and with a radius equal to the radius of the guide-pulley, cut o^6 1 at o^6, and cut D^h M^h extended at o^5. Points o^6, o^4, and o^5 will be in a straight line, and the line joining them will be the edge view of the central circle of the guide-pulley. About this line o^6 o^5 draw a rectangle as shown, the width of the rectangle being made equal to the width of the face of the guide-pulley. Through o^4 draw a line perpendicular to $o^6 . o^5$, which will be the revolved position of the center line of the guide-pulley shaft. The method of revolving back to get the axes of the ellipses, which are the projections of the edges of the pulley in its actual position, and to get the projection of the shaft, will be clear from a careful study of the figure.

The vertical projection of the guide-pulley is found by revolving over in exactly the same way, the distance O^v o^7 being equal to the perpendicular distance of O^h from the ground line. It is well to assume a definite length for the shaft, whether this be the actual length which the shaft would have or not. The length assumed in the figure is T^1 T^2 (same as T^3 T^4), and half of this is laid off each side of o^4 in the revolved horizontal projection.

This completes the two elevations of the guide-pulley. The plan is drawn as follows: Find the projection L L of the two ends of the shaft as shown by the construction lines; then revolve over by drawing line K K^1 at any convenient place parallel to L L, drawing perpendiculars through the two points L meeting the parallel line at K and K^1, and laying off on these perpendiculars the distances K^1 L^1 and K L^2, equal respectively to L^D P^D and L^E P^E in the elevation. The line L^1 L^2, joining the points L^1 and L^2 thus found, is the revolved position of the shaft, and should be equal in length to T^1 T^2 and T^3 T^4 in the elevations.

We can now draw the rectangle which represents the revolved position of the guide-pulley at the middle of the line L^1 L^2, and find the ellipses from this rectangle in the same way as we found the ellipses from the rectangle in the other two views.

The belt is drawn in accordance with the same kind of reasoning as was used in determining the way the belt would look in the other kinds of quarter-twist belts which we have studied.

Belts Connecting Shafts in Same Plane but not Parallel. It very often happens that a belt must connect two shafts which are on the same level, but which are not parallel. The connection can be made, whatever the angle between the shafts, by the use of two guide-pulleys. If the two main shaft pulleys are of the same diameter, the belt may be made to run in either direction by putting both guide-pulleys—or *mule pulleys* as they are often called—on the same shaft, which will be perpendicular to the plane containing the axes of the main shafts. That is, if the main shafts are horizontal, the shaft for the guide-pulleys will be vertical. If the main pulleys are of different diameters, the guide-pulleys may still be placed on the same vertical shaft, but in this case the belt can run in only one direction. If the belt is to run in either direction, the guide-pulleys are placed on separate shafts, which are usually adjustable in position, so that they may be tipped at the proper angle to receive and deliver the belt.

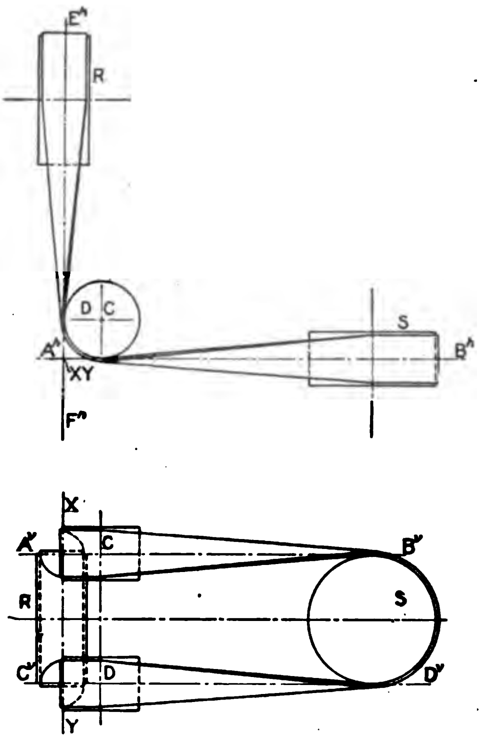

Fig. 53. Belts Connecting Two Shafts not Parallel, but in Same Plane.

We shall first take the case where the two main pulleys are of the same size. Fig. 53 shows the arrangement, the upper view being the plan and the lower view the elevation. R and S are the two main pulleys, and C and D are the two guide-pulleys. The line X Y is the line of intersection of the planes of the main pulleys. The location of the guide-pulleys is sufficiently clear from the drawing without further explanation.

Fig. 54 shows the arrangement when the main pulleys are of different diameters. Here the pulleys can turn only in the direction shown by the arrows; for if the direction were reversed, the belt would leave the pulleys. The pulleys in plan appear the same as in Fig. 53. In the elevation the upper guide-pulley, which receives the belt from S, has its central plane tangent to the pulley S at the point where the belt leaves S, as shown by the line $A^v B^v$. The lower guide-pulley, which receives the belt from R, has its center plane tangent to R, as shown by $C^v D^v$. If the pulleys were to turn in the opposite direction, the upper guide-pulley would have its plane tangent to R, and the lower one, its plane tangent to S.

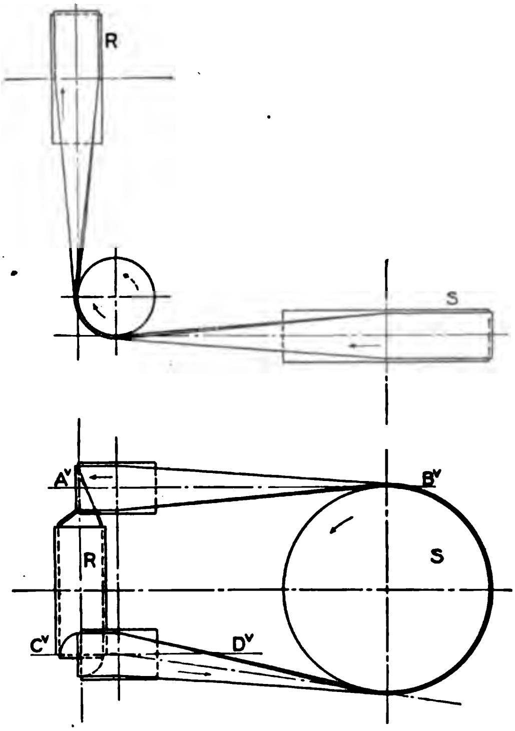

Fig. 54. Belts Connecting Two Shafts not Parallel, but in Same Plane.

Cone Pulleys. It is often necessary to provide a range of speed variation in a shaft belted from a line shaft running at a constant rate. A familiar case of this kind is the ordinary lathe spindle. This may be done by shifting the belt from end to end of either a pair of cones or conoids, Fig. 55, depending on whether the belt is crossed or open. To work satisfactorily, a shipper must be located at each cone or conoid to guide the belt; otherwise it may climb. As these shippers give trouble and wear the belt, and the belt itself is unequally stretched, it is usual to approximate the conoids by stepped cones, or cone pulleys. In order to have the belt equally tight for each pair of steps, the diameters of the steps must be so proportioned that the length of belt remains constant.

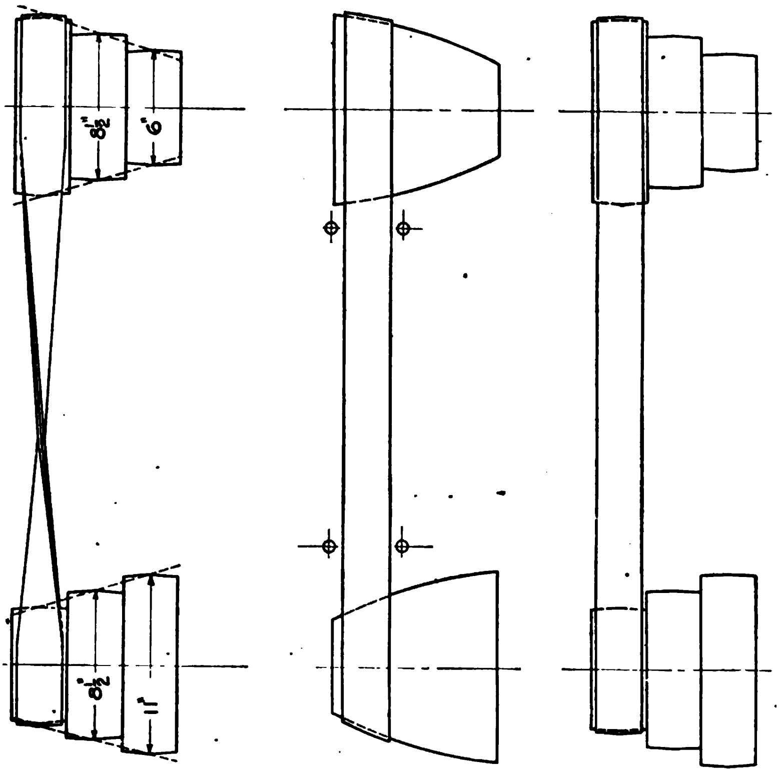

Fig. 55. Cone Pulleys.

It can be shown, geometrically, that for a crossed belt this condition is obtained when the sum of the diameters of each pair of steps is constant. In Fig. 55, a three-step cone with crossed belt is shown; and, adding together the diameters of the pair of steps connected by the belt, we have 11 + 6 = 17 for the constant of this cone pulley.

For an open belt, an extended calculation is necessary for diameters giving a constant belt length, and a simple graphic method for laying out the cones has been published by Mr. C. A. Smith in *Transactions of the American Society of Mechanical Engineers* (Vol. 10, p. 269). Here the distance between shafts and diameters for one cone pulley is assumed, or is known from the conditions of the drive.

In Fig. 56, lay off A B equal to the distance between centers of shafts; and with these points as centers, draw circles C and D equal respectively to the maximum and minimum diameters of the given cone pulleys. Draw the belt line E F. From a point G, half-way between A and B, erect G H perpendicular to A B, and make it equal to 3.1416 A B (for the sake of space, it is not drawn to this scale in the cut). With H as a center, draw a circle tangent to E F; then the belt line of any other pair of pulleys must be tangent to this latter circle.

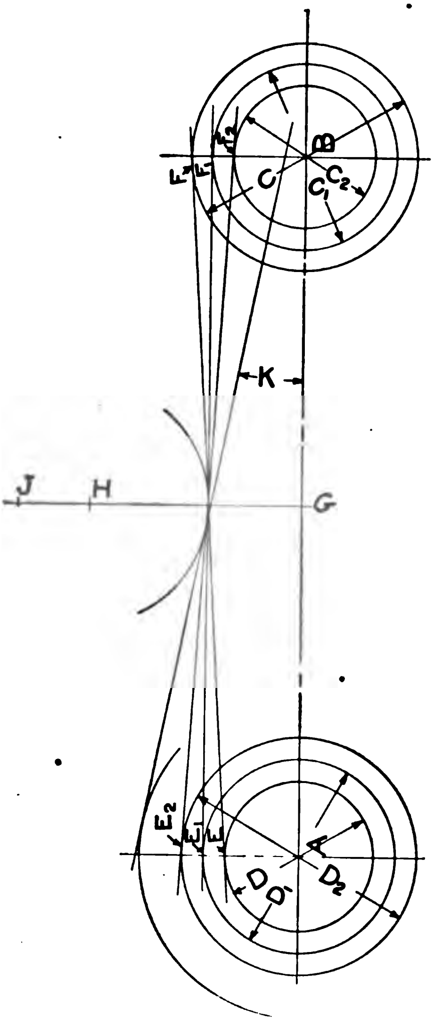

Fig. 56. Lay-out of Cone Pulleys.

Assume D_1 the diameter of one of the pulleys, and draw a common tangent to circles D_1 and H, producing it past the center B. From B draw a perpendicular B F_1 to the common tangent; and with B F_1 as a radius, draw in the circle C_1, which will give the pulley required to work with D_1, and having approximately the same length of belt as on pulleys D and C. Continue the process until the required number of steps have been obtained. It must be noted that the limit for which this construction can be used is reached when the belt angle K is equal to 18 degrees. When the angle K is between 18° and 30°, proceed as follows:

Locate another point J on the line G H so that the distance H J is equal to .298A B; draw a tangent to circle H, making an angle of 18° to the line of centers A B^1; and from point J draw an arc tangent to this tangent. Make all belt lines which are greater than 18° tangent to this arc.

Belt Holes. Very often a belt has to pass through a floor or partition. The holes through which the belt runs should be large enough to be sure that the belt shall never strike the sides, but it is desirable that they should be no larger than is necessary to accomplish that result. Accordingly, the holes should be laid out so that they may be cut in the right place and at the proper angle. Figs. 57 to 59 show the method of getting the position of the belt holes for the various kinds of belts. The top of the floor only is shown, as it is all that would need to be used in drawing the belt holes, the holes being cut through the floor parallel to the center line of the belt.

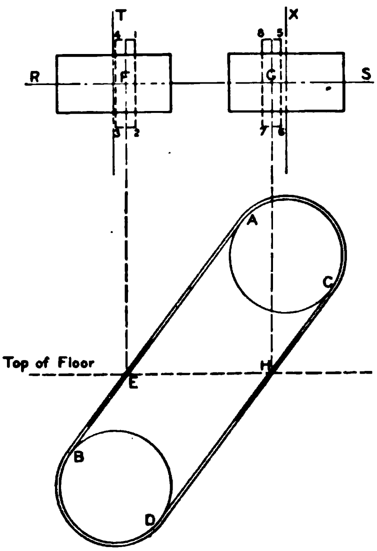

Fig. 57. Cutting Holes in Floor for Common Open Belt.

In Fig. 57 we have a common open belt. The circles representing the pulleys are drawn, and the belt drawn around them. A short piece of the pitch-line is also drawn in each part of the belt where it passes through the floor. These parts of the pitch-line are simply lines parallel to, and half-way between, the lines which represent the outer and inner faces of the belt. Next draw the two rectangles which represent the plan view of the pulleys, and draw through them the center line R S. From the points E and H, where the pitch-line intersects the line representing the top of the floor, draw perpendiculars to R S, meeting it in points F and G. F and G are the center points of the rectangles 1 2 3 4 and 5 6 7 8, which form the outline of the belt holes on the surface of the floor. The long dimension of the rectangles will be parallel to the shafts on which the pulleys are located. After the belt holes are so found, the distances of their center lines to the right or left of the lines T and X respectively (which are the center lines of the shafts), can be measured on the drawing, and the workman can mark them out on the floor by plumbing down (or up) from the shafts, getting the lines T and X on the floor directly under or over the center of the shafts, and thus locating on the floor the points F and G, and consequently the belt holes, from the dimensions taken from the drawing.

Fig. 58 shows how to draw the holes for a crossed belt. Draw the two views of the pulleys and the center lines A C and D B of the belt in the elevation; also the center line R S in the plan. It is well, also, to draw the belt complete in the elevation, as it makes it easier to determine which way the belt holes will slant. From points E and H, where the center lines of the belt intersect the floor line, draw E L and H K perpendicular to R S, and meeting R S in F and G. The points F and G are the center points of the belt holes, and it only remains to determine the angles which the center lines of the holes make with T and X respectively. These will be the same as the angles made with H K and E L. When the belt is leaving the pulley at A, a line drawn perpendicularly across to its inner face would occupy the position indicated by the dotted line *a a′* in plan; and the belt, in passing from A to C, twists through an angle of 180 degrees, and the line which was at *a a′* will occupy the position *c c′*. Therefore, when the belt has passed from A to H, it will have twisted through an angle which will bear the same relation to 180 degrees that the distance A H bears to the distance A C. That is, if A H = ¼ of A C, the angle J G K is ¼ of 180 degrees, or 45 degrees. Whether the angle J G K shall be laid off to the right or to the left of line H K, must be reasoned out by considering which way the belt twists in passing A to C. The angle of the other belt hole (L F M) is determined in the same way.

Fig. 59 shows the method of finding the belt holes of a plain quarter-twist belt, similar to Fig. 47. The centers G and F in plan

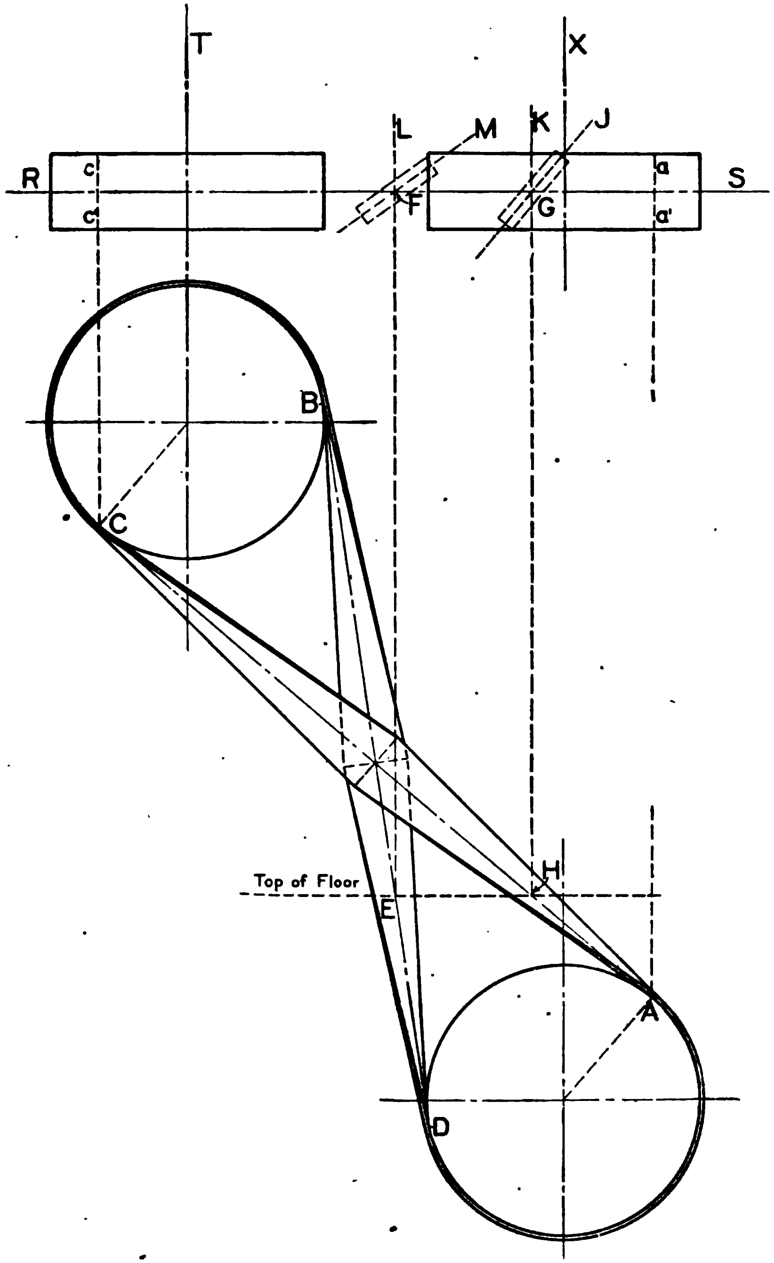

Fig. 5‹. Cutting Holes in Floor for Crossed Belt.

are found by projecting from the elevation, as shown by the construction lines. The angle which the center line of hole at G makes with the center line of the shaft, is found by dividing 90 degrees in

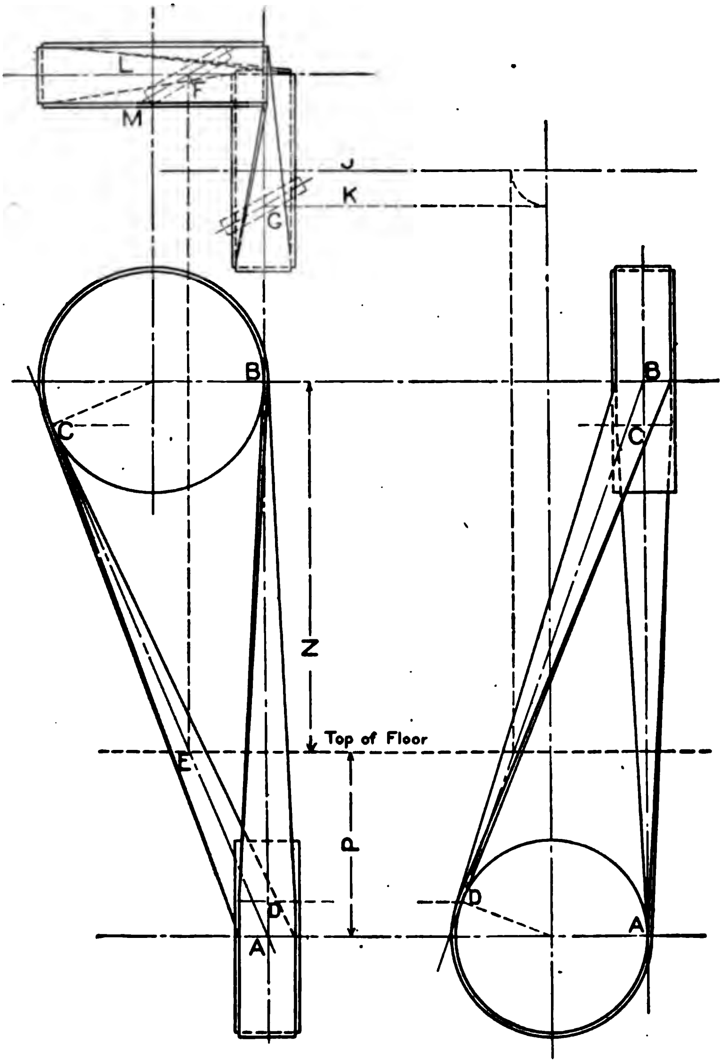

Fig. 59. Cutting Holes in Floor for Quarter-Twist Belt.

the ratio of the distances P and N. The angle of the center line of the belt hole at F with the center line of the shaft, is found in a similar manner, by dividing 90 degrees in the ratio of the distance A E to E C. It is usually sufficiently accurate, however, after having found the angle at G, to draw the center line of the other hole parallel to it.

General Notes. A belt drive is working under the most favorable conditions when, though not pulled up excessively, the belt "hugs" the pulleys tightly and wraps a large proportion of their circumference.

In the case of two pulleys of different diameters, made of the same material, connected by a belt, the belt will slip first on the smaller pulley, partly because the wrap of the belt is less upon that one, partly because the belt does not "hug" so tightly, owing to the smaller radius of curvature to which the belt must be bent. Sometimes the smaller pulley is lagged with rubber or leather so as to give it increased grip on the belt, thus making up for the tendency to slip, due to its small diameter.

On an inclined or horizontal drive the slack side should be on top, and the tight or pulling side underneath, as the weight and slackness of the belt will act together to cause it to sag and increase the wrap. As shown in Machine Design, Part II, the greater the arc of contact, the greater the driving force which may be obtained from the belt.

Vertical drives should be avoided as much as possible, as here the weight of the belt is always tending to decrease the "hug" on the lower pulley.

Increasing the diameter of pulleys, the same linear speed of belt being maintained, does not increase the power transmitted, except by permitting the belt to "hug" the pulleys more tightly; and the larger the pulleys, the better this condition becomes, providing we do not exceed a certain economic speed for the belt. Flexible link belting, made of small leather links joined together by steel wire, gives excellent results, especially when used on horizontal drives, but it is rather expensive to install. A pulley rim perforated with small holes, to prevent any air cushion beneath the belt, is another means of increasing the "hug."

Belt-adjusting devices are often provided for changing the distances between the pulleys, thus enabling the proper tightness to be always maintained. Motor and dynamo bases are provided with slides and set screws for such adjustment.

A tightener pulley is often used to increase the wrap of the belt or maintain the proper tightness. This is an idler pulley, which is weighted, or adjusted by screws against the belt. While such a pulley is a very ready and simple means of accomplishing the purpose, yet it should be remembered that the shaft carrying it is subjected to heavy pressure in its bearings, due to the belt tension; and the friction of the drive is considerably increased thereby. Tightener pulleys are used only when specific conditions prevent the results from being otherwise secured.

It is generally preferable to use belts of two or more thicknesses for the sake of side stiffness, and also in order that any local imperfections of the leather in one layer may be taken care of by the other. Where the belts are to be shifted laterally, stiffness is an important item. If too pliable a belt is used on cone pulleys, the edges are apt to curl up, and the belt tends to climb and chafe against the side of the step, twisting like a corkscrew, and sometimes jumping from one step to another.

A good distance between the centers of shafting for ordinary belt drives is from 20 to 25 feet. With greater distances, the belt is apt to flop and run in waves; while at a less distance, the necessary tightness of the belt results in undue stretching. In crossed belts, the above distances should be especially adhered to; for, with a wide, stiff belt and a short distance between centers, there is an excessive amount of rubbing on the sides of the belt, as well as strain caused by the twisting.

It is well to use as few crossed and quarter-twist belts as conditions will permit. In quarter-twist belts, the side angle, where the belt leaves the pulley, should be kept under 25°, as considerable power is lost in side slipping. For the least distance between the shafts a safe value is obtained if the distance is made not less than 2½ times the diameter of the larger pulley. A narrow belt gives better results than a wide one on twisted belting.

TOOTHED GEARING

Fig. 60 represents a pair of discs fastened to shafts A and B respectively and touching at the point P. If these discs be pressed tightly against each other, sufficient friction will be produced between them to cause one to drive the other. The number of revolutions B would make in a given time, would be to the number of revolutions made by A as A P is to B P; or,

$$\frac{\text{Revolutions B}}{\text{Revolutions A}} = \frac{\text{A P}}{\text{B P}}.$$

Such friction discs will transmit but very little power without slipping; and even when required to transmit small power, cannot be depended upon to drive *positively*, as the least wear or loss of adjustment is liable to make them slip. Hence teeth are provided on each disc, such that they will lock together and make it sure that when one disc is rotated the other must move also, without regard to whether the discs are pressed tightly together or not. In fact, it is desirable that this side pressure be avoided, in order to prevent excessive friction in the bearings of shafts A and B.

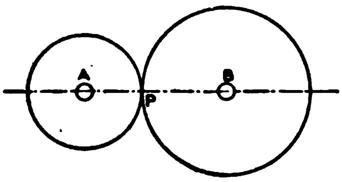

Fig. 60.

Any shapes whatsoever of teeth would answer, provided they interlocked, so far as positive driving is concerned. But in order that the *revolutions of the shafts shall always be inversely as the contact radii*, or

$$\frac{\text{Revolutions B}}{\text{Revolutions A}} = \frac{\text{A P}}{\text{B P}},$$

it can be shown by geometry that the *common normal drawn through the point of contact of any pair of teeth must always pass through the point* P.

A pair of gears, therefore, may be considered to be based on two discs, touching as in Fig. 60, and provided with teeth such that these two conditions are fulfilled:

1. Positive driving at all times.
2. The common normal through the point of contact of any pair of teeth always passing through the pitch point.

The circles corresponding to the discs are known as *pitch circles*, their diameters *pitch diameters*, and the point of contact P the *pitch* *point* (see Fig. 61). The distance, measured radially, from the pitch circle to the top of the tooth is called the *addendum*; and the circle through the top of the tooth, the *addendum circle*. The distance, measured radially, from the pitch circle to the beginning of the fillet at the bottom of the tooth, is called the *dedendum*; and the circle through this point the *dedendum circle*. In order that the top of the tooth on one gear shall not strike the surface between the bottoms of the teeth on the other, a further distance is allowed between the dedendum circle and the root circle, known as the *clearance*. The distance from the center of one tooth to the center of the next, measured on the pitch circle, is called the *circular pitch*, and is evidently equal to the circumference of the pitch circle divided by the number of teeth.

In order to run together, two gears must have the same circular pitch. The number of teeth in a pair of gears is proportional to the circumference of the pitch circles, and therefore to the pitch diameters, or pitch radii. The speeds of the shafts carrying the gears, being inversely proportional to the diameters of the pitch circles, are also inversely as the numbers of teeth.

Since the circular pitch is equal to the circumference of the pitch circle divided by the number of teeth, there is a fixed relation, for any given gear, between the pitch diameter and the number of teeth. This relation is known as *diametral pitch*. Diametral pitch is not a distance, like circular pitch, but is *the number of teeth per inch of pitch diameter* of the gear. For example, if the diameter of the pitch circle of a gear of 60 teeth were 20 inches, the number of teeth per inch of diameter would be $\frac{60}{20} = 3$, and the gear would be described as a "60-tooth, 3 diametral-pitch gear." The product of the circular pitch times the diametral pitch, is always equal to the constant, 3.1416; that is, if we have one kind of pitch, and wish to change to the other, we divide 3.1416 by the given pitch. For example, 4 diametral pitch is equal to $\frac{3.1416}{4} = .7854''$ circular pitch. Again, 2″ circular pitch is equal to $\frac{3.1416}{2} = 1.57$ diametral pitch. Note carefully that diametral pitch is not "inches," *but number of teeth per inch of diameter.*

Diametral pitch is very convenient to use, as the calculation is simpler than with circular pitch, and the pitch diameters of the gears come in even figures, or in even fractions of the pitch. For machine-cut gears it is universal practice to use diametral pitch in the specification. For cast gears, where the teeth are fashioned by the pattern maker, it is common to use circular pitch.

The thickness of the tooth L M, Fig. 61, is practically the same as the space T L for machine-cut gears. For cast teeth, however, the tooth must be thinner than the space, to allow for the inaccuracies of the pattern and casting. This allowance measured on the pitch circle is called *backlash.*

These terms are illustrated in Fig. 61; also the *common normal* K P to a pair of teeth in contact. Gear A, being the driver in the direction shown, a pair of teeth are in contact at point K. The curves of the teeth being of the correct shape, if a common tangent be drawn, and a perpendicular erected at the point of tangency K, it will pass through the pitch point P. Now, as the gears move in the direction of the arrows, the teeth slide upon each other, and the point of contact changes, coming closer and closer to point P, then passing through P, and going on reaches some point as J, which, in the present example, represents the second pair of teeth in contact. During all this motion of the teeth, the common normal at every point of contact will pass through the pitch point P, thus fulfilling the condition of uniform velocity as stated on page 72.

It will be remembered that the pressure line between two surfaces, as illustrated in the discussion of cams, is the common normal at the point of contact. Now, a pair of gear teeth is like a cam and its follower; and if we wish to find the direction of the pressure between them, we simply draw the common normal. Hence, knowing that with the teeth of proper outline the common normal will pass through the pitch point, we merely find the point of contact of any pair of teeth and connect it by a straight line to the pitch point, thus giving the direction of pressure between the teeth at the given position.

Two kinds of curves fulfill the requirement for gear teeth, that the common normal shall pass through the pitch point. These are the *cycloidal* and *involute* curves. The latter curve, for many reasons, has almost entirely displaced the former. The cycloidal

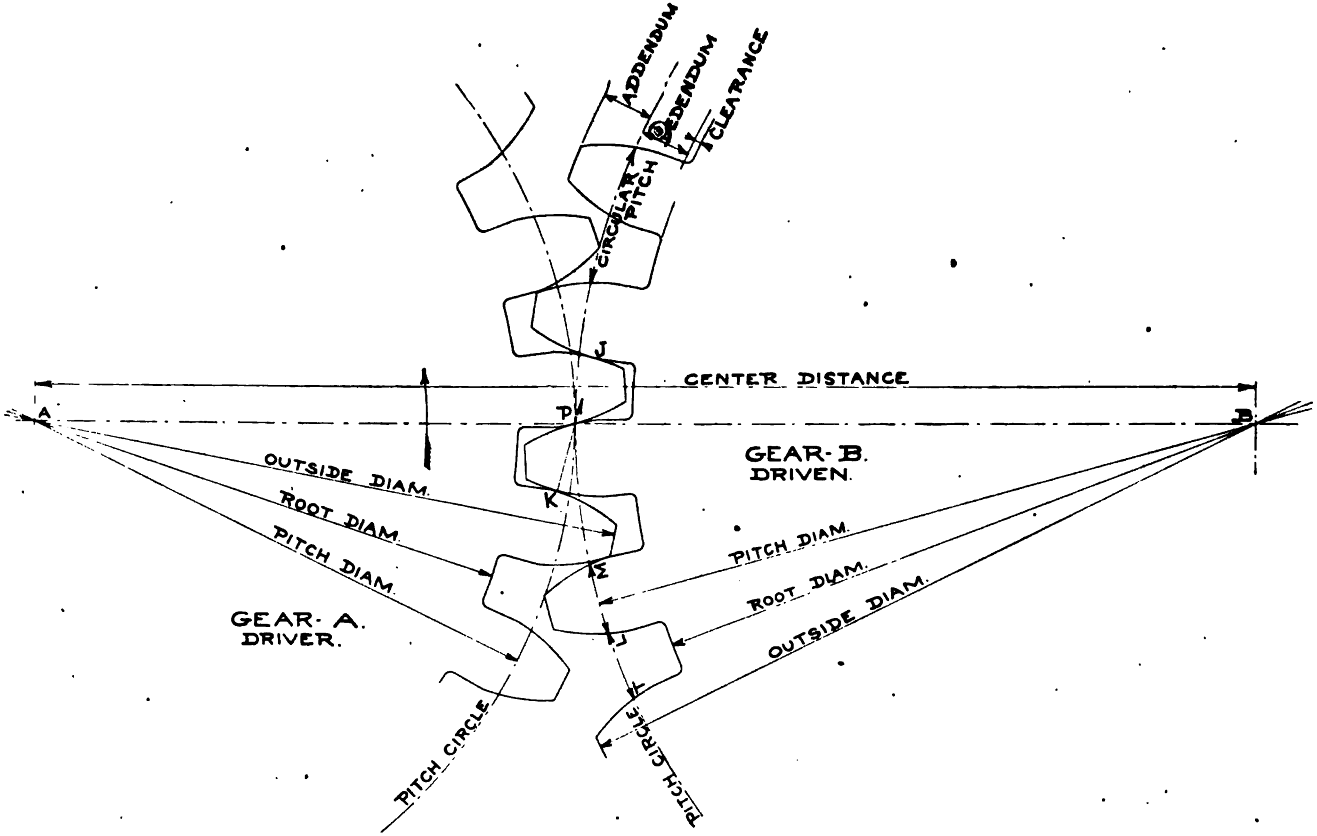

Fig. 61. Lay-out of Gears.

curve is useful in special cases, and is still adhered to by its few advocates, as having peculiar merit, even for standard work. The general and best standard practice, however, is unalterably committed to the involute system, and experience has shown the reasons therefor to be sound ones.

The student can best approach the subject of the design of gear teeth through a study of the cycloidal system, the principles being capable of clearer illustration. Hence this system will be first presented.

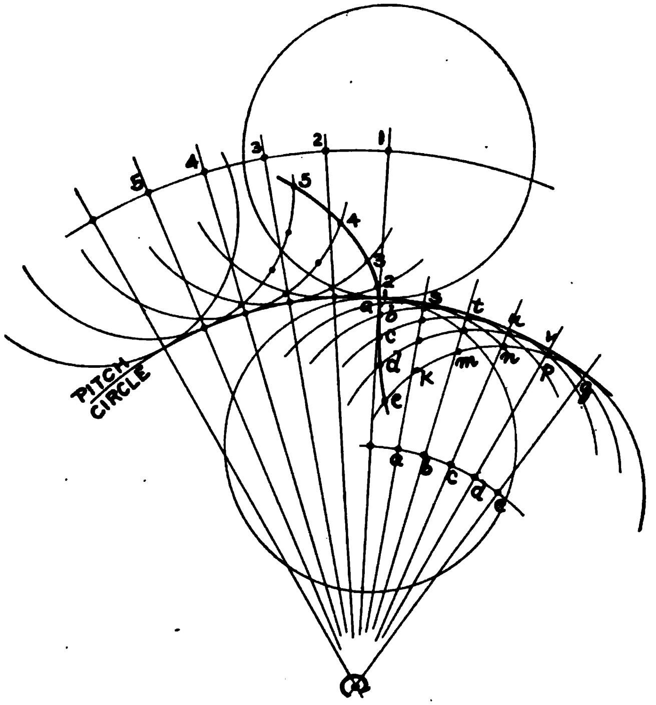

Fig. 62. Cycloidal Curves.

Cycloidal Spur Gears. The method of drawing the cycloidal curves by the use of rolling circles is illustrated in Fig. 62. The accurate curve, having been developed, may be transferred by the tracing-cloth method, as in cams, to each individual tooth; or arcs may be found by trial which approximate to the true curve; or a templet of stiff paper or cardboard may be made.

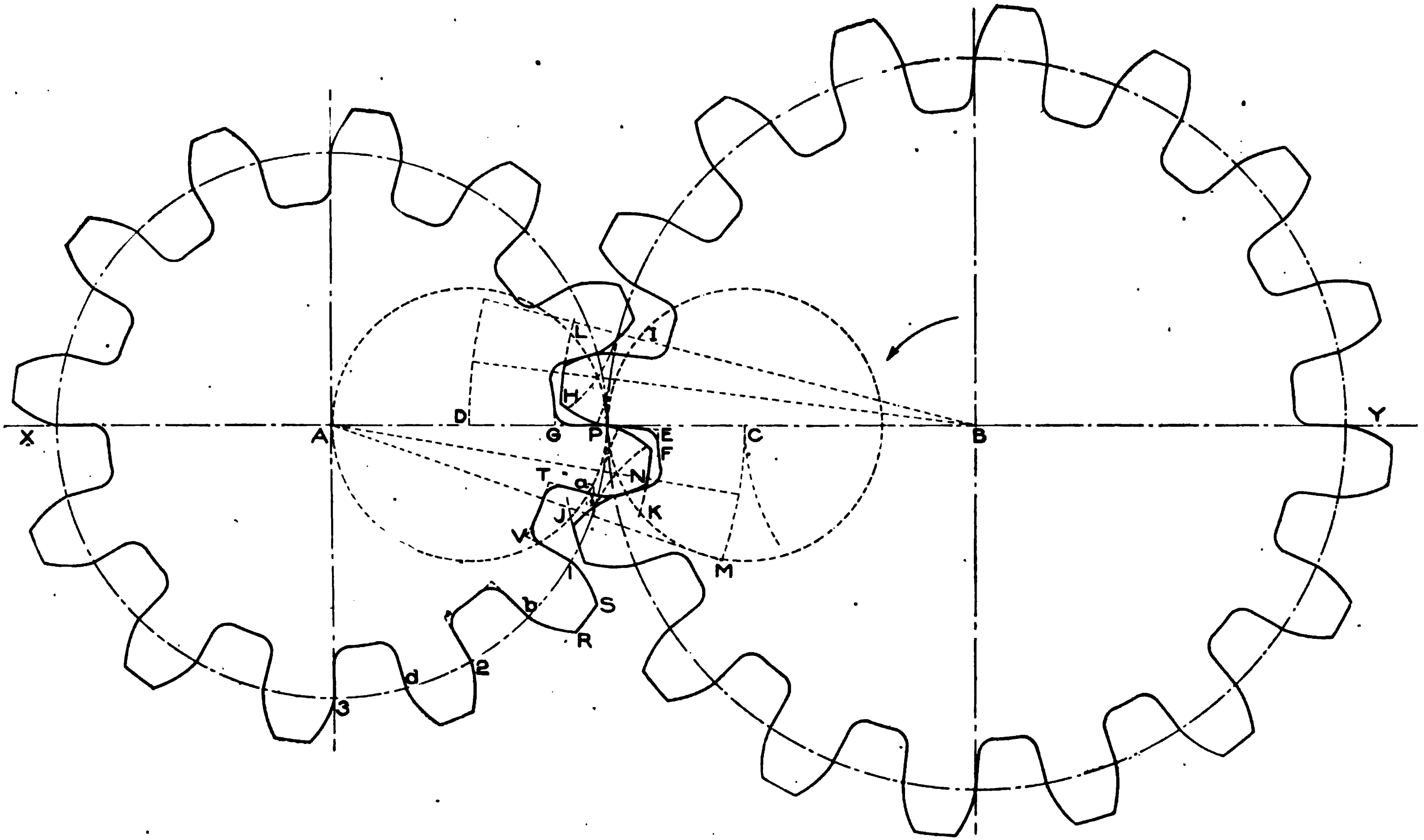

Fig. 68. Pair of Epicycloidal Gears.

Fig. 63. shows a pair of epicycloidal gears designed to run together. The centers of the gears are at B and A; the pitch circles are shown in dot and dash, and are in contact at the pitch point P. The circle whose center is C, shown dotted, by rolling on the inside of the pitch circle of the gear B, generates the hypocycloid P E, which forms the flanks of the teeth on gear B; and by rolling on the outside of pitch circle of gear A, generates the epicycloid P F, which forms the faces of the teeth on gear A. In like manner the circle whose center is D, by rolling on the inside of the pitch circle of gear A, generates the hypocycloid P G, which forms the flanks of the teeth on A; and by rolling on the outside of the pitch circle of B, generates the epicycloid P H, which forms the faces of the teeth on B. The circles C and D are called the *describing circles.* If the gear B is the driver and is turning in the direction shown by the arrow, the flanks of its teeth act on the faces of the teeth on A from the point where they first come in contact until the point of contact reaches the pitch point; and from the pitch point on until the contact ceases, the faces of the teeth of B act on the flanks of the teeth of A. In other words, the hypocycloidal part of the tooth curve on one gear is generated by the same describing circle that generates the epicycloidal part of the tooth on the other gear with which it is in contact. This must always hold true, in order to have the gears run properly. The arc I P of the describing circle C, together with the arc J P of the decribing circle D, forms what is called the *path of contact*; that is, the point of contact between the teeth is always somewhere on the line I P J. If the gear A were the driver, the direction of rotation remaining the same, the path of contact would be L P K.

To design a pair of epicycloidal spur gears we must have given the pitch (either diametral or circular), the diameters of the pitch circles, or the number of teeth, and something to determine the size of the describing circles. Manufacturers have found by experience what are the best ratios of describing circles to pitch circles, and gears are designed according to those ratios. It is not well to have the diameter of the describing circle greater than ⅝ the diameter of the pitch circle, and it is better to have it smaller. If a set of gears is to be made, any one of which is likely to run with any other one, the same size describing circle must be used for the faces and flanks of all the gears; and this describing circle is often taken ⅝ the diameter of the smallest gear of the set. Sometimes when two gears are not part of an interchangeable set, but are designed to run with each other only, the diameter for the describing circle for the flanks of each gear is made equal to the radius of that gear; and when this is the case the flanks are radial straight lines; or, as it is usually stated, the gears have *radial flanks.*

In Fig. 63, the two describing circles are of the same size and equal to the radius of the smaller gear, thus giving radial flanks on this gear. Let us proceed with the design of this pair of gears, given dimensions as follows: Gears to be 4 pitch (that is, as explained previously, four teeth per inch of pitch diameter); gear A to have 12 teeth; gear B, 16 teeth; addendum equal to the diametral pitch; clearance equal to $\frac{1}{8}$ the addendum; describing circles each equal to radius of gear A. The steps in the process of drawing the gears are as follows:

First.—Calculate the diameters of the pitch circles.

Second.—Draw the center line X Y on the paper; and on this center line locate the centers A and B a distance apart equal to $\frac{1}{2}$ the sum of the two pitch diameters. About these centers draw the pitch circles, of diameters as calculated. This will make the pitch circles tangent at the pitch point P.

Third.—Calculate the addendum and dedendum, adding this amount to and subtracting from the radii of the pitch circles. Then draw the addendum and dedendum circles with the radii thus found.

Fourth.—Draw the root circles with radii equal to the radii of the pitch circles minus an amount equal to the dedendum plus the clearance.

Fifth.—Draw the describing circles tangent to each other and to the pitch circles at the point P.

Sixth.—With the describing circle C rolling on the outside of the pitch circle of A, generate the epicycloid P F, continuing it until it meets the addendum circle of A. With the describing circle D rolling on the inside of the pitch circle of A, on the *opposite* side of line of centers from which the circle C rolled, generate the hypocycloid P G. Since the diameter of D is equal to the radius of the pitch circle of A, the hypocycloid P G will be a radial line; and consequently, after the student has become familiar with this fact, it will not be necessary actually to roll the circle to generate such a hypocycloid. The epicycloid P F and the hypocycloid P G together form one side of the tooth of gear A.

Seventh.—Divide the circumference of the pitch circle into as many equal parts as the gear has teeth, and through these points draw curves like the curve G P F. This may be done by making a templet of stiff paper that will just fit the curve G P F, and by means of this templet transferring the curve to the points 1, 2, 3, etc. Next find the points *a*, *b*, *d*, etc., half-way between 1 and 2, 2 and 3, etc., since there is to be no backlash, and through these points draw curves similar to G P F, but turned so as to curve the other way. Now, by filling in with full lines that part of the addendum circle between the points F and N, R and S, etc., and filling in the root circle between T and V, etc., we have the outline of the teeth on the gear A. In practice, instead of making sharp corners at T and V, as shown by the dotted lines, fillets are put in with arcs of circles, these fillets being made as large as possible and still allowing space so that the corner of the teeth on the other gear shall not strike.

Eighth.—Construct the teeth on the gear B in the same way as the teeth on A were constructed, the describing circle D generating the epicycloid P H by rolling on the outside of the pitch circle of B, and the describing circle C generating the hypocycloid P E by rolling on the inside of the pitch circle of B. The hypocycloid is not a straight line in this case, as the diameter of C is not equal to the radius of the pitch circle of B.

The calculations for the above case are as follows: 4 pitch means 4 teeth per inch of diameter; and as there are 16 teeth in B, its diameter will be $\frac{16}{4} = 4$ inches; 12 teeth in A will give $\frac{12}{4} = 3$ inches diameter.

The addendum for a standard machine-cut gear is usually made equal to the dedendum, and a distance equal to the reciprocal of the pitch. Hence, to find the addendum and dedendum in the present case, take the reciprocal of 4, which is $\frac{1}{4}$ inch.

The clearance, being $\frac{1}{8}$ the addendum, is equal to $\frac{1}{8}$ of $\frac{1}{4} = \frac{1}{32}$ inch.

If the student tries to follow the above description by actually drawing these gears, it will be found necessary to draw them to about 3 times their actual size in order to bring out the points clearly.

That is to say, the pitch circles should be made 9 and 12 inches; the addendum and dedendum $\frac{3}{4}$ inch; the clearance $\frac{3}{32}$ inch; the numbers of teeth, of course, remaining 12 and 16.

ANNULAR GEARS

An *annular gear* is a ring with teeth on the inside of it. Fig. 64 shows such a gear, with center at A, meshing with its pinion. The method of drawing such a pair of gears is similar to that just described for two spur gears. Here the circle C, by rolling on the *inside* of the pitch circle of A, generates the *faces* of the teeth of A; and the circle D, by rolling on the *outside* of the pitch circle of A, generates the *flanks* of the teeth on A. The shape of the teeth of the pinion

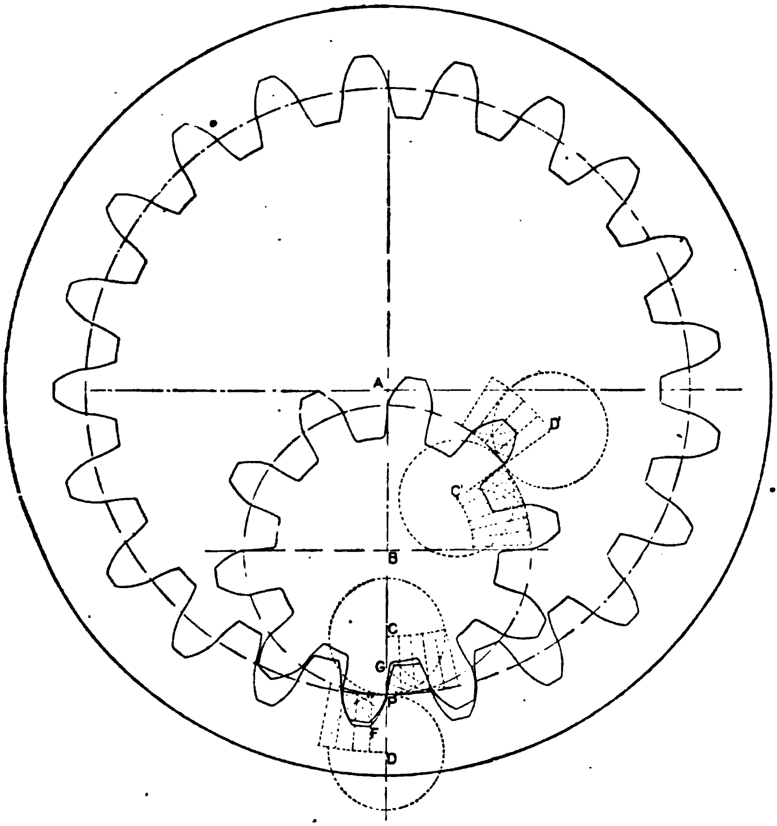

Fig. 64. Annular Gear.

in the figure is found with the same describing circles placed at C′ and D′ to avoid mixing up the lines.

In designing an annular gear and pinion the diameter of the gear must never be so small that the distance from center A to center B shall be less than the sum of the radii of the two describing circles. If this should be the case, the teeth would interfere with each other.

RACK AND PINION

A *rack* is a gear whose pitch line is a straight line instead of a circle. Fig. 65 shows an epicycloidal rack in gear with a 16-tooth pinion. The describing circles are of the same size in the figure, although they might be of different sizes. The teeth on the pinion are drawn as described for Fig. 63, the construction lines for drawing them not being shown. The curves which form the faces and flanks cf the rack are cycloids. The describing circle C, rolling on the pitch-line of the rack, generates the cycloid P F, which forms the flanks of the rack teeth; and the describing circle D, rolling on the pitch-line of the rack, generates the cycloid P H, which forms the faces of the rack teeth. The addendum and root lines are drawn parallel to the pitch-line, and at a distance from it equal respectively to the distances A and R of the pinion. The teeth are spaced off on the pitch-line of the rack by laying off the distances P K, K L, etc., equal to the circular pitch of the pinion.

INVOLUTE GEARS

We have seen in the preceding pages how the outlines of cycloidal gear teeth are generated by a point in a circle rolling on the pitch-line. We have noted that the point of contact between the teeth is always somewhere on the describing circles, drawn tangent at the pitch point. The outlines of *involute* gear teeth, which are far more common than cycloidal teeth, are generated by a somewhat similar process. In the case of the involute, however, the describing point is located on a straight line, rolling, not on the pitch circle, but on another circle inside the pitch-line, known as the *base circle*. The result of rolling a straight line, as noted above, is the same as if we stand up on the drawing board a small cylinder of diameter equal to the base circle, fasten one end of a string to some point in its circumference, and then allow the string to unwrap from the cylinder, a pencil point at the free end of the string marking on the paper below it the involute curve.

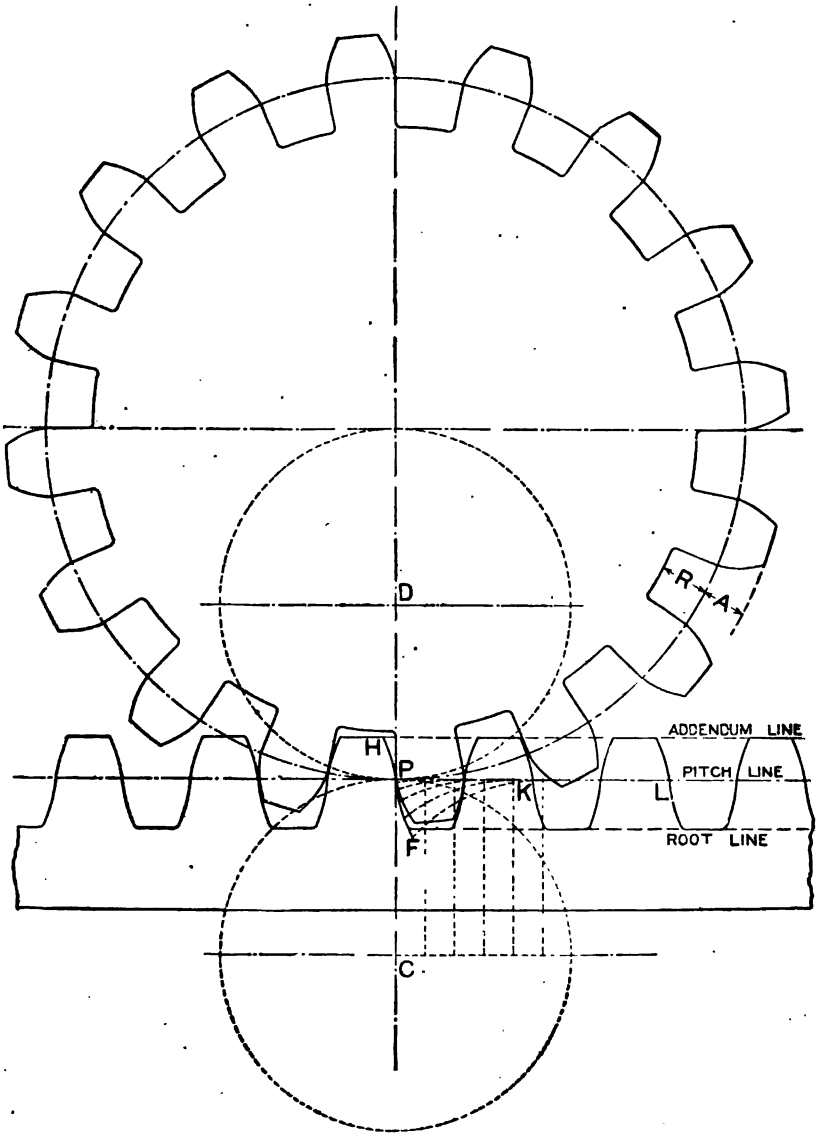

Fig. 65. Rack and Pinion.

The method of drawing the involute curve is shown in Fig. 66; and it is obvious from this figure that the curve can never extend inside the base circle, although it may go any distance above it.

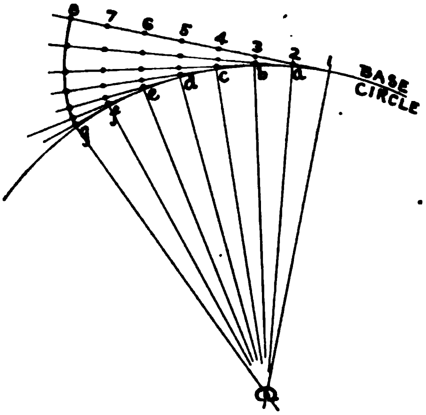

Fig. 66. Involute Curve.

Fig. 67 shows a pair of gears with involute teeth, drawn according to the principles stated below. The circular pitch and diameters of pitch circles are calculated in the same way as described for cycloidal gears. The centers A and B are chosen, and the pitch circles drawn tangent at the pitch point P, as before. In involute gears, the point of contact between the teeth is always somewhere on an inclined line, C D, passing through the pitch point. The angle which this line makes with the tangent X Y, is called the *angle of obliquity*. The size of this angle has an important bearing on the action of the gear teeth; and there are special conditions which, for the best results of tooth action, would call for widely different angles. It is not well, however, to have the angle of obliquity of different values, as it would then be impossible for any two gears to run together, except those based on the same angle. The angle of obliquity which has been quite generally adopted and which seems to fulfil the average conditions best, is 15°. In the present case, therefore, draw the line C D at an angle of 15° with the tangent X Y; with A and B as centers, draw circles tangent to C D; these circles are called the *base circles*. The addendum, dedendum, and root circles are then drawn at the same relative distance from the pitch circles as in the case of cycloidal gears. The spacing of the teeth is now accomplished by stepping the dividers, set to the circular pitch, around the pitch circle. At any convenient points on the base circle, as G and E, generate the involutes in accordance with the method of Fig. 66, or as explained in Part II. Then, by the tracing-cloth method, or by the use of a templet fitted to this curve, draw in the tooth curves at points R,S,T, etc., on the pitch circles. This gives us the working part of the teeth, and the remainder of the tooth to the root circle consists of a radial line. Fillets are put in at the bottom of the teeth, as in the case of cycloidal gears.

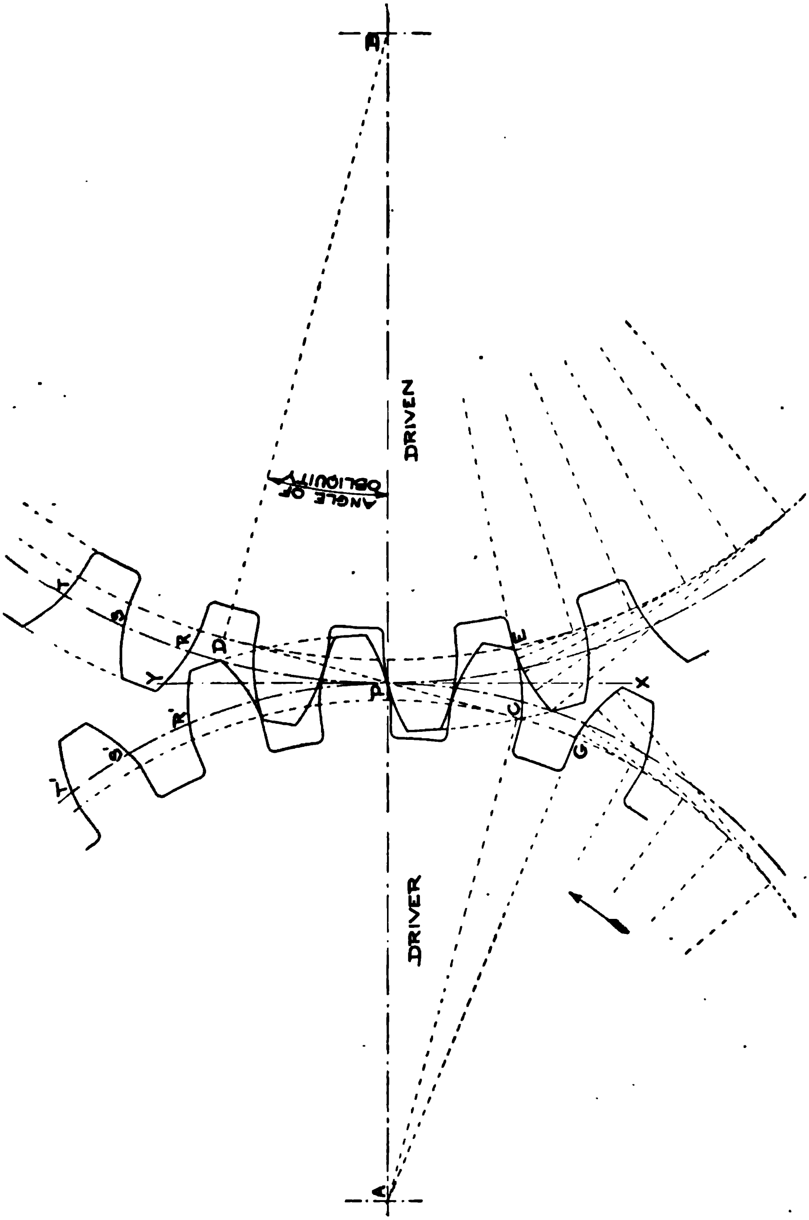

Fig. 67. Pair of Gears with Involute Teeth.

As has been stated above, the point of contact between the teeth is always somewhere on the line C D; it is therefore obvious, that, if the circle struck through the top of the tooth on one gear cuts the base circle of the other gear at a point outside of point C, there can be no true contact at the top of the tooth. Instead of there being true contact, the top of the tooth will actually dig into the lower portion of the tooth of the other gear. This is known as *interference*, and is overcome by slightly rounding off the top of the tooth down to the circle through point C, so that it will clear. Since the path of the point of contact is along the line C D, this line also represents the common normal to any pair of teeth in contact, and therefore is the *line of pressure* between the teeth. The obliquity of this line of pressure to the line of centers A B causes a thrust between these centers, tending to force the gears apart; and this has been considered an objection to the use of involute gears. With the standard 15° involute, however, experience has shown that this thrust is ordinarily of small importance. A similar thrust exists in cycloidal gears, but is constantly changing in value, being a maximum at the beginning and end of contact of a pair of teeth, and zero when the pair of teeth are in contact at the pitch point. It will be noted that the involute tooth is of simpler outline than the cycloidal, being a single curve instead of a reverse curve. If the exact distance between the centers A and B of a pair of involute gears be not maintained, owing to wear or to some other cause, the gears will still continue to run perfectly together; whereas in the case of cycloidal gears the action is seriously impaired by such a condition.

BEVEL GEARS

Bevel gears are used to connect shafts whose axes intersect. The angle between the shafts is not necessarily a right angle, but this is the most common angle used. Fig. 68 shows a pair of bevel gears connecting two shafts whose axes intersect at a right angle.

The cones O P A and O P B are called *pitch cones*; the cones C P B and D P A, *normal cones*, and it is on these normal cones that the outlines of the teeth are laid out; B P and A P are the *pitch diameters* of the gears, and are found from the pitch and number of teeth just as the pitch diameters of spur gears are found.

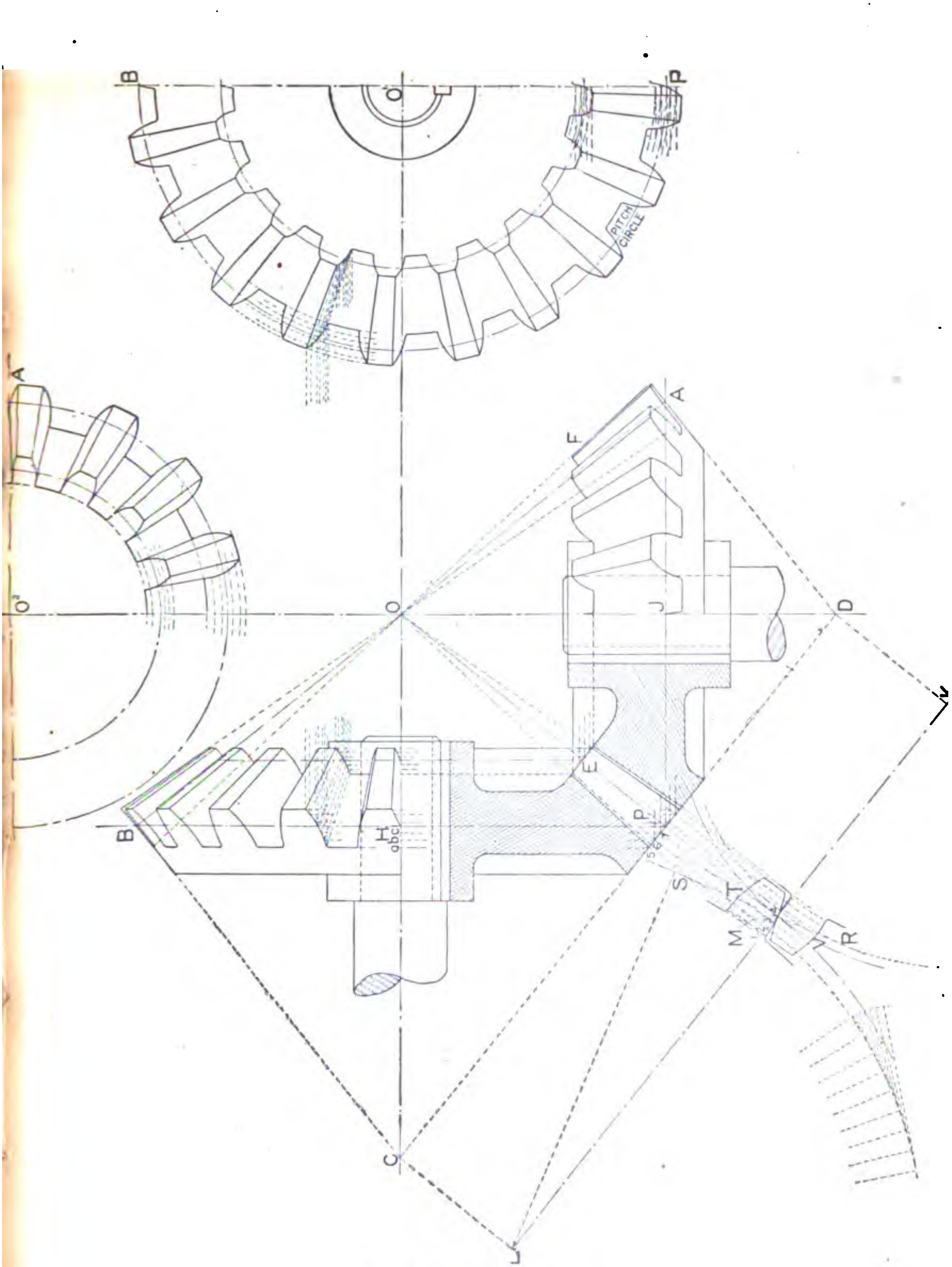

Fig. 68. Pair of Bevel Gears.

To draw such a pair of gears, we must have given the angle between the shafts, the pitch and number of teeth in each gear, and the face of the tooth P E. The outlines of the teeth may be either involute or cycloidal; the addendum, dedendum, and clearance are determined by the same empirical rules as were applied to the other gears which have been discussed.

Referring to Fig. 68, the gears shown are 2-pitch, 16 and 20 teeth respectively, with face P E equal to 2 inches.

According to previous understanding, the addendum or the dedendum for a standard tooth is the reciprocal of the diametral pitch—or, in this case, $\frac{1}{2}$ inch. Making the clearance $\frac{1}{8}$ of the addendum, would give $\frac{1}{8}$ of $\frac{1}{2} = \frac{1}{16}$ inch. The teeth are of the involute form, with an angle of obliquity of 15°. Choosing point O, draw the lines O C and O D, making an angle of 90° with each other; calculate the pitch diameters of the gears; lay off on O C the distance O H, equal to $\frac{1}{2}$ the pitch diameter of the smaller gear; and through H draw a line perpendicular to O C. In like manner lay off on O D the distance O J, equal to $\frac{1}{2}$ the pitch diameter of the larger gear; through J draw a line perpendicular to O D, meeting the perpendicular which is drawn through H at P; and make H B equal to H P, and J A equal to J P. From A, P, and B, draw lines to O, producing the pitch cones; through P draw C D perpendicular to O P, meeting O C and O D in C and D respectively. Join C B and D A, and we have the normal cones. Through C, P, and D, draw perpendiculars. Draw L M K parallel to C P D at any convenient distance. Draw arcs of circles tangent at the point M. These arcs are now to be treated as pitch circles on which to design the tooth curves, in exactly similar fashion to the method already outlined for spur gears.

Through point M draw the line of obliquity S R, and draw the base circles tangent to this line. With the addendum chosen as above, equal to $\frac{1}{2}$ inch, it will be found that the addendum circle of the larger gear will cut the line of obliquity beyond the point R, where S R is tangent to the base circle of the pinion. This means that true contact cannot occur at the top of the gear tooth, so the tooth should be slightly rounded off, to prevent interference with the flank of the pinion. The limit of this rounding-off of the point of the tooth is determined by striking a circle with center L through the point R, as it is obvious that below this point on the tooth of the gear there will be true involute contact. The root circles are drawn by setting off the clearance, as in the preceding cases. One tooth on each gear is drawn on the development of the pitch circle, and the completion of the drawing of the teeth in the several views of the gears is merely a problem in projection.

With L as a center, draw a series of arcs (shown dotted) cutting the tooth which was drawn on the pitch circle, and the line L M K, at 2, 3, 4, etc.; from 2, 3, 4, etc., draw lines perpendicular to C D, cutting C D at 5, 6, 7, etc.; from these points draw lines to O; along P O lay off P E; through E draw a line perpendicular to P O, cutting 5 O, 6 O, etc.; and from the points of intersection draw other lines parallel to P B. With center O′, taken at any convenient place on C O prolonged, and with radii equal in turn to *a* 5, *b* 6, etc., draw circles as shown. On the circle which is drawn with H P as a radius (marked "pitch circle"), space off the circular pitch; and on each of the circles in turn, lay off the teeth of the same width as they are on the corresponding circles drawn through 1, 2, 3, 4, etc. The rest of the construction can be understood by a careful study of the figure. The other gear is drawn in the same way.

The drawing of the teeth for bevel gears whose shafts intersect at another angle than a right angle, is accomplished by following out the same principles as noted in the case at hand. The skeleton outline of such a pair of gears is shown in Fig. 69, the angle between the axes being 60°. These gears are 2-pitch, 16 and 20 teeth respectively, the same as in the previous case; and their construction affords an interesting comparison therewith.

To draw the teeth on a pair of bevel gears as described in Fig. 68, is a tedious process and requires considerable patience and drafting skill. It is really little more than an exercise in advanced projection drawing, but, as such, is valuable to the student. It must not be thought, however, that to detail a pair of bevel gears for manufacture, such a drawing is necessary. Usually, standard proportions of teeth are specified, and the detail of the gears is comparatively simple. An illustration of a pair of bevel gears of standard proportions of teeth, detailed ready for the workman's use, is shown in Fig. 39, Part IV, and it is seldom necessary to show more.

General Remarks On Gear Teeth. The foregoing study of the outlines of gear teeth is given in brief and elementary form. The student cannot hope to gain a familiar comprehension of the action going on between the teeth of gears, without going more deeply into the subject than is possible in these pages. The action of gear teeth is one of the most complicated subjects to investigate and understand, as with each new condition of number and type of teeth, new points of action are developed.

A good practical treatise on gear teeth is "A Treatise on Gear Wheels" by George B. Grant; and the student is referred to this book for a further study of the subject.

There are many special points to be observed in designing the outlines of gear teeth, in order to insure the best operation of the gears. These points cannot be well explained without the actual undertaking of the design of the teeth. If the student wishes to familiarize himself with tooth action, he cannot do better than to choose a variety of cases, and lay out each one studying the several points as they come up.

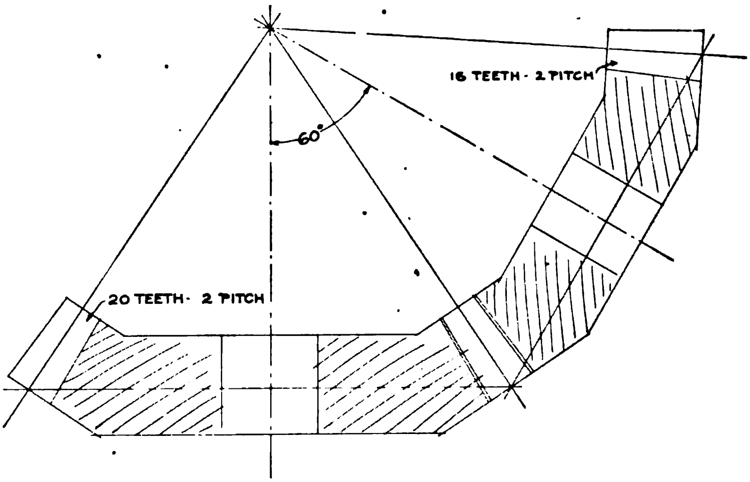

Fig. 69. Bevel Gears with Axes at Oblique Angle.

It should be remembered that the action of a *small* pinion, meshing into a *large* gear is considerably different from that of two large gears meshing into each other. With certain relative numbers of teeth of gear and pinion, as many as three pair of teeth may be in contact at all times; while in certain other combinations, but two are in contact at all times, and in certain others only one. Changes in the tooth dimensions, diameters of describing circles, angles of obliquity, etc., alter all these conditions, so that there are an endless variety of combinations, each of which presents some new feature only to be understood by actual lay-out of the particular case.

In gear-tooth work, the student will often find it an advantage to make the lay-outs to double the actual size, and sometimes larger. A fine, hard pencil must be used, and extreme accuracy in determining the points must be adhered to. The lay-out of gear teeth is one of the severest tests of the draftsman's ability in line work.

PLATES.

Plates XX to XXIV A inclusive, are to be drawn by the student for practice in applying the principles of screws, springs, cams, belts, and gears. These plates are to be made the same size as those of the preceding Instruction Papers of the course; viz., 11 inches by 15 inches, outside, with a margin of ½ inch, making the clear space for the drawing 10 inches by 14 inches.

PLATE XX.

Fig. 1. Make a theoretical drawing of a right-hand, single, sharp V-thread screw, 4 inches in diameter, 6 inches long, 1 inch pitch. Construct the true helix as shown in Fig. 12 of the Instruction Paper. Show dotted construction lines for one complete turn of the helix.

Fig. 2. Make a conventional representation of a single, right-hand, square-thread screw, 2 inches in diameter, 3½ inches long, ½ inch pitch, (2 threads per inch).

Fig. 3. Make a conventional representation of a single, right-hand, U. S. standard V-thread screw, 2 inches in diameter, 3½ inches long, 4½ threads per inch. Use form shown in Fig. 24.

Fig. 4. Make a conventional representation of a single, right-hand, sharp V-thread screw, 1 inch in diameter, 3½ inches long, 8 threads per inch. Use form shown in Fig. 23.

Fig. 5. Same as Fig. 4 except make left-hand, and double thread.

Fig. 6. Draw a right-hand helical spring, $\frac{5}{16}$-inch round wire, inside diameter of spring 1⅜ inches, length 3½ inches, pitch ⅝ inch, using form shown in Fig. 8.

All the figures on this plate are to be made and inked in on the drawing paper.

PLATE XXI.

Fig. 1. Draw a plate cam, the top turning to the right, to raise a slide (having a roll at its lower end) 1½ inches with harmonic motion, while the cam turns through 135°, then to allow a fall with uniform motion while the cam turns through 180°, and rest for the remaining 45°. Make the center line of motion of the slide pass through the axis of the cam, the roll 1 inch in diameter, and at its lowest position 1½ inches above the axis of the cam. Show one view only, and do not show the shaft or hub of the cam, these being merely explanatory on the plate as shown. Use a hard pencil, with a fine point, and show pressure lines.

Fig. 2. Draw a plate cam, the top turning to the left, to raise a roll, similar to that of Fig. 1, 1½ inches with uniform motion, while the cam turns through 150°, then to rest for 30°, then to fall for the remaining 180° with a uniformly accelerated and retarded motion. Make the center line of motion 1 inch to the right of the axis of the cam, and at its lowest position the center of the roll 1½ inches (measured radially) from the center of the cam. Show pressure lines.

As inking will impair the accuracy of both of the figures on this plate, it should be handed in just as the pencil work leaves it. Special care must be taken to have the lines fine, sharp, accurate, and clear. Accuracy is of first importance in this case, and though the plate should be cleaned up as much as possible without destroying the construction lines, yet it cannot be expected that the plate will be as clean when finished as though it were inked, thus giving opportunity to rub over the whole drawing.

PLATE XXII.

This shows plan and one elevation of two pulleys located on two shafts at right angles to each other. Draw these two views and another elevation, showing the belt on the pulleys in all three views, and place

SCREWS
AND
SPRINGS
EXACT AND CONVENTIONAL
REPRESENTATIONS

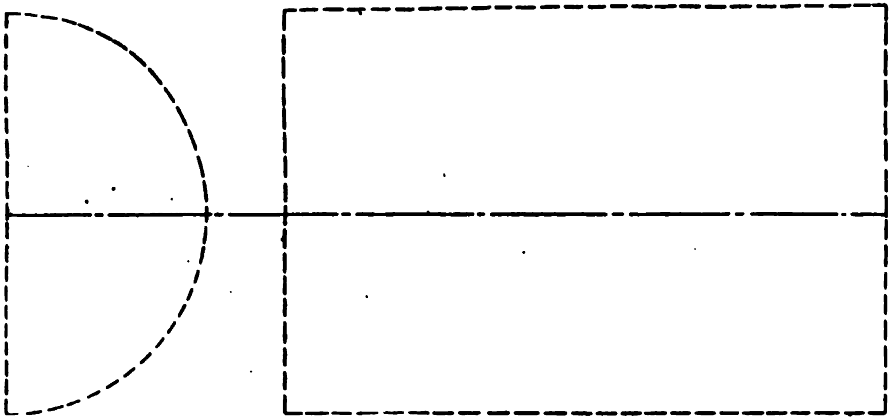

FIG 1

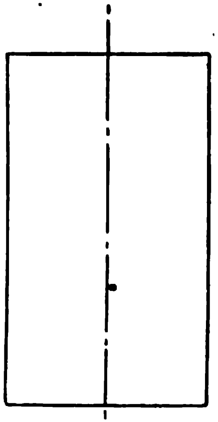

FIG. 2

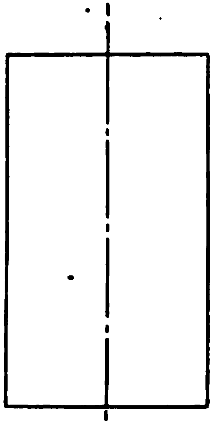

FIG. 3

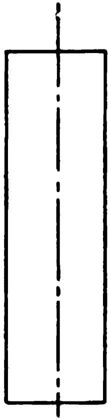

FIG 4

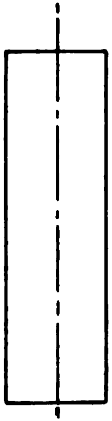

FIG. 5

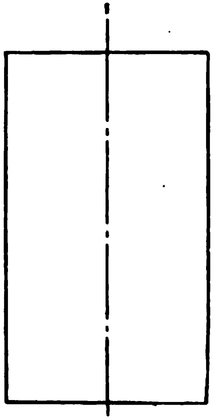

FIG. 6

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PLATE XXI

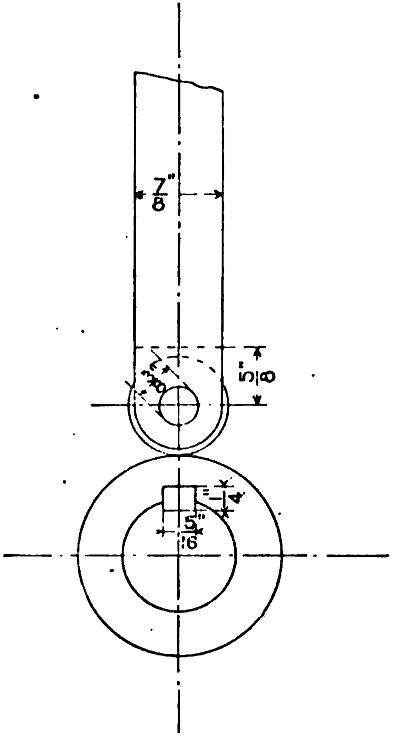


FIG. 1

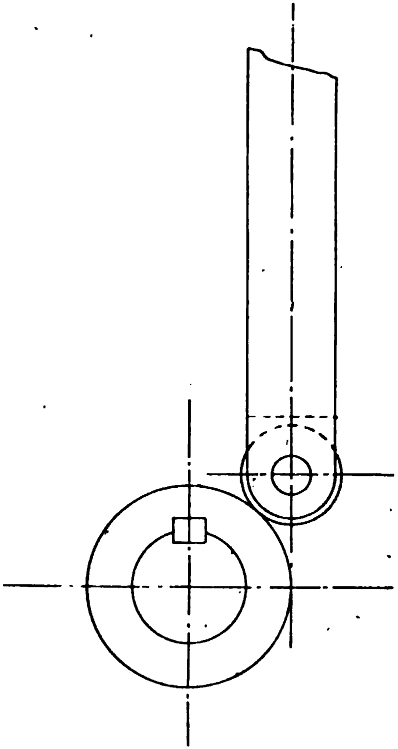


FIG. 2

arrows to show the direction of motion of the pulleys in order that the belt may stay on. The dimensions are given in feet and inches, but the drawing cannot be made full size. As stated in the title, the scale may be 1 inch per foot, that is, the distance between the shafts being 8 feet, it is made on the drawing 8 inches. Other dimensions are to be reduced in the same proportion, or to one-twelfth size. This plate should be inked in on the drawing paper.

PLATE XXIII.

Fig. 1. With center 3 inches from the extreme left of the sheet, and on a horizontal line through its middle, strike the arc of a pitch circle with 6 inches radius. Take describing circles of 4 inches diameter, and roll them upon both sides of this pitch circle, producing at least three epicycloidal teeth, diametral pitch 1½, addendum ⅔ inch, dedendum ⅔ inch, clearance ⅛ inch, thickness of tooth equal to one-half the *circular* pitch. Note that the *diametral* pitch, 1½ as given, is not "inches," but number of teeth per inch of diameter of pitch circle. In order to lay out the teeth, the circular pitch must be calculated as explained in the Instruction Paper.

Fig. 2. With center 5½ inches from the extreme right of the sheet, and on the same horizontal line as before, strike the arc of a pitch circle with 6 inches radius. Take angle of obliquity equal to 15°, and draw the base circle for involute teeth. Draw at least three full involute teeth, diametral pitch 1½, addendum ⅔ inch, dedendum ⅔ inch, clearance ⅛ inch, thickness of tooth equal to one-half the circular pitch. As before, the *circular* pitch must be calculated, and when the teeth are finished it should be noted how different is the shape of the involute tooth from the epicycloidal, yet both are correct working teeth for the same diameter of pitch circle.

Both figures of this plate should be handed in just as the pencil work leaves it, for the same reasons as for Plate XXI.

PLATE XXIV.

Draw a horizontal line through the middle of the sheet, and, choosing a center one inch from the left-hand side of the sheet for the pinion, draw an 18-tooth pinion meshing with a 30-tooth gear of the involute, spur-gear type. Make teeth 2 diametral pitch, angle of obliquity 20°, addendum ½ inch, dedendum ½ inch, clearance 1/16 inch, thickness of tooth one-half the circular pitch. Draw as many teeth as there is room for across the plate, show the length of the path of contact, and state the greatest number of teeth which can be in action at any one time.

This plate should be inked in with as fine lines as possible, in order not to greatly impair the accuracy of the work.

PLATE XXIVA.

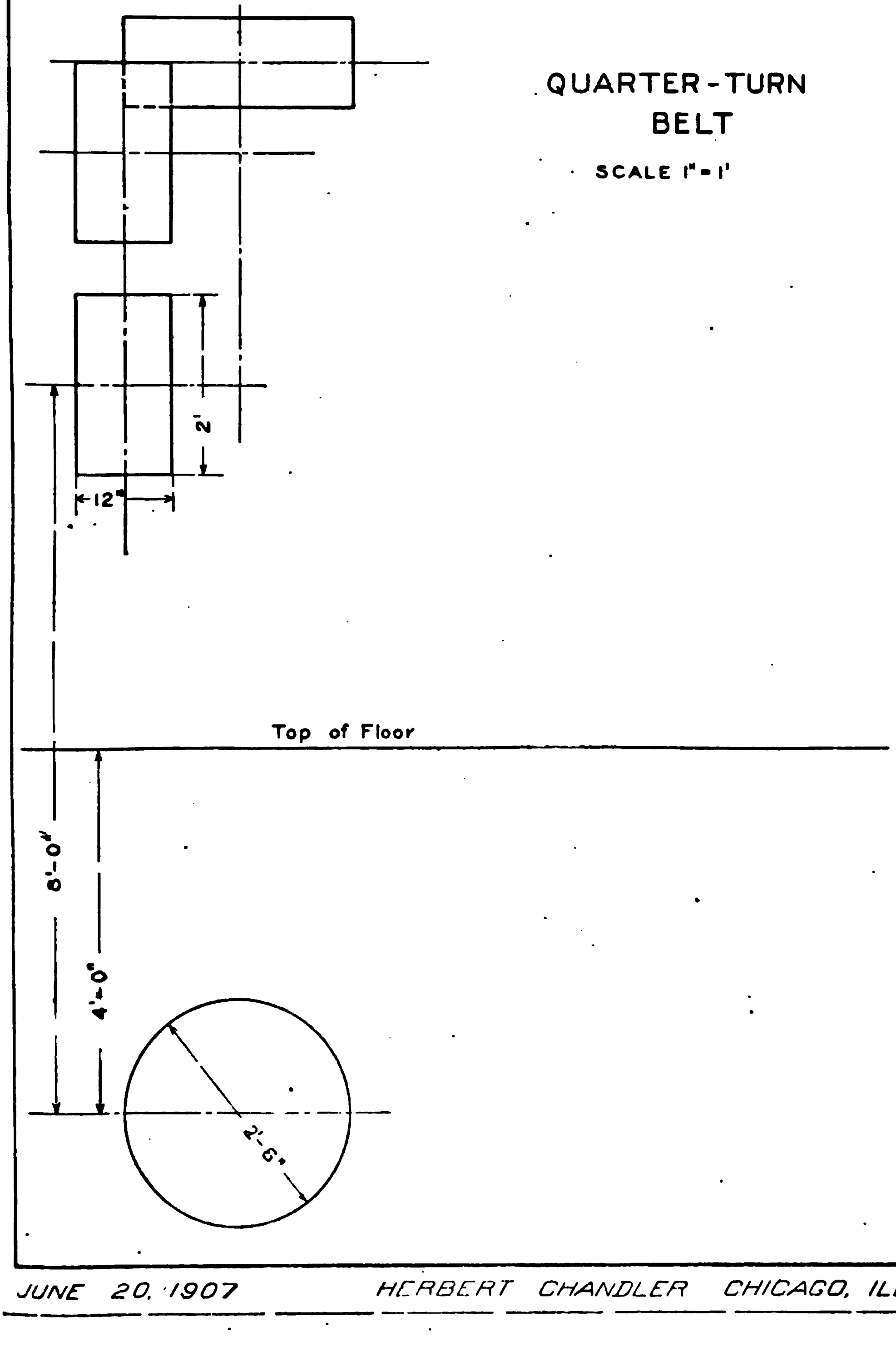
Draw a horizontal line 5¾ inches from the bottom of the sheet and a vertical line 4 inches from the left-hand side as axes for a 27-tooth bevel gear and a 16-tooth pinion respectively. Make the teeth 4 diametral pitch, addendum ¼ inch, dedendum ¼ inch, clearance $\frac{1}{32}$ inch, length of face 1½ inches. Make an outline like Fig. 69, indicating the pitch diameters, pitch cones, and normal cones.

This plate should be inked in, fine lines being used for the construction lines, and heavy ones for the body of the gears, which may be made any proportions to suit the student's conception of what they should be. This may be treated as a case for development of the student's mechanical judgment and sense of proportion, based, of course, on the exact skeleton outline made from the data as given.

PLATE XXII

QUARTER-TURN
BELT

SCALE 1" = 1'



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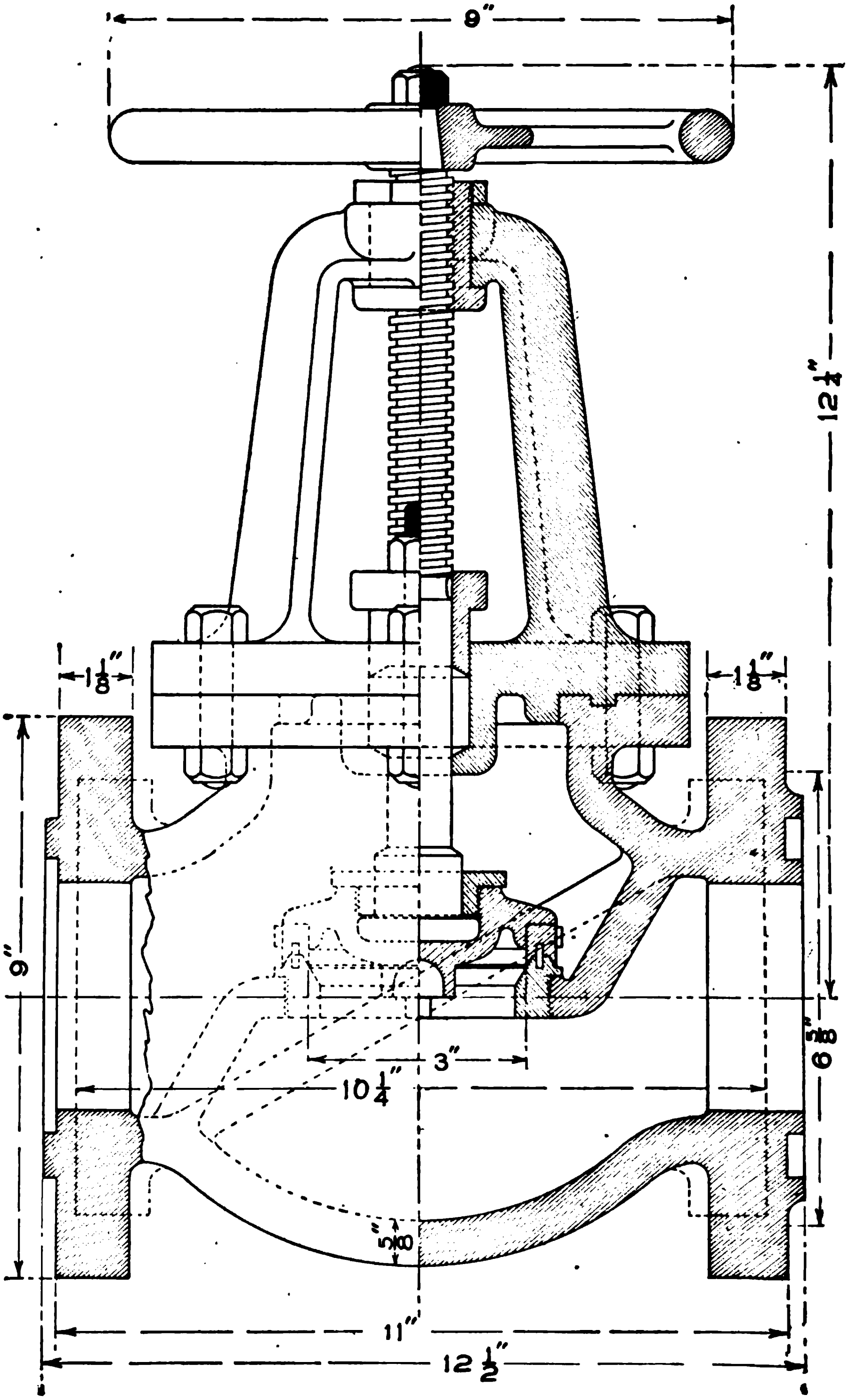


WORKING DRAWING SHOWING PRINCIPAL DIMENSIONS OF SPRING SEAT VALVE.

Crosby Steam Gage & Valve Co.

MECHANICAL DRAWING.

PART VI.

WORKING SHOP DRAWINGS.

In Parts I to IV, inclusive, the fundamental principles of Mechanical Drawing were explained and illustrated. The production of working drawings has also been discussed to some extent, and the usual characters and symbols explained and applied. The elementary work already outlined has been treated chiefly from the standpoint of correctness of line representation considered by itself, without a detailed study of the use to which the drawings so produced are to be applied.

Evidently this is the proper method, for the student should gain a thorough understanding of the principles which underlie line representation before attempting to apply them to any extended practical use. In all of this preceding work it is intended that the theoretical principles shall overshadow any incidental references made to practical application, however true and pertinent the latter may be for purposes of illustration. Hence, before taking up any advanced work, the student should fully realize the importance, in fact, the absolute necessity, of thoroughly understanding the fundamental principles which have been outlined in the books which have preceded Part VI.

At this point the student must realize that a lack of proper elementary and fundamental training will make him "go lame" at every point of his course, and probably prevent the attainment of proficiency which otherwise would naturally and almost instinctively come with advanced study. It is the thorough and ready knowledge, always at his fingers' ends, of *all* the principles of Mechanical Drawing, which makes the expert draftsman.

Plan and Scope of Advanced Work. It is now intended to throw an entirely different light on the matter, and view the subject of Mechanical Drawing from a purely practical standpoint; viz., that of Utility. It is assumed that the student understands and can use the principles which have been previously discussed.

If in a working shop drawing we choose to modify any of these theoretical principles, it will be because of increased value in *Utility* of the drawing. For example, we may desire to omit some portions of an elevation or plan or side view of a complicated casting, because certain details will thus be more clearly brought out. We may make a "zigzag" section to show construction which, by absolute fidelity to theoretical principle, would be confused, or hidden in a maze of dotted lines. We may find it convenient to place in some unoccupied corner of a drawing a layout which could not be in the least justified by any rule of projection. A multitude of transgressions like these occur on good drawings, and they are certainly justifiable from the standpoint of *Utility*, which is the true ultimate end sought for in a practical shop drawing.

These variations from the theoretical are not strictly conventionalities, because they are not classified or established, so far as we know, but are the spontaneous outgrowth, as the occasion demands, of the draftsman's purpose to make his drawing one of greatest *Utility*. He can, however, safely transgress a principle only when he thoroughly knows the principle; otherwise a blind deviation from the theoretical path will inevitably lead to difficulty.

All of the above is intended to impress the student with the idea that theoretical principles are his best, in fact, his only tools to work with; but they are not "self-hardening," like "mushet" steel; they are like the finest grade of tool steel, which must be tempered and ground and used with the best judgment of the operator, to secure the most satisfactory results.

Student Drawings. A student's early drawings are usually unsatisfactory, even to himself. Somehow they do not look like those seen in shops, and as a rule he is unable to see why this is so. Of course the difference is to some extent due to the experience of the professional draftsman. However, the superior results of the latter's work are attained largely through his systematic and workmanlike habits of execution. It should encourage the student in his early attempts to know that these essentials to the infusion of life and shop spirit into a drawing can be analyzed, outlined and grasped at the outset by earnest, intelligent effort, and really good workmanlike results obtained. To discuss, and if possible to impart these essentials of a working shop drawing to the student, is the purpose of the present book.

Essentials. The two chief essentials of a shop drawing, under which general heads a multitude of detail requirements can be summed up, are:

(1.) Absolutely complete and definite instructions from designer to workman.

(2.) Least possible cost in dollars and cents of production of the drawing measured by the draftsman's time.

It makes no difference how much we may attempt to disguise these two elements, the fact will still be apparent that "*complete instructions furnished for the least money*" is what the manufacturing shop is after, and what will be assumed as a basis for judgment as to highest commercial utility.

Completeness of Drawings. As to the first point, that of completeness and definiteness of instruction, there must be no question of degree. If the information which the drawing furnishes is positive and complete, the drawing is good. If doubt arises in the workman's mind as to what the designer intended by a certain line or dimension, or if the dimension be omitted, the drawing is bad. There is no middle ground. The instructions are either present or absent, and the drawing good or bad accordingly.

The workman of to-day is not permitted to assume dimensions or shape. It is his business to execute the draftsman's orders; it is, however, often his privilege to choose his own way of doing it, but further than this modern practice does not allow him to go. He is held as rigidly to the orders specified by the drawing as the locomotive engineer is held to his bit of tissue telegraphic order to proceed, without which he dare not enter the next block. The drawing is supreme; it is official; it must be plain, direct and all-sufficient. It is the draftsman's business to make it thus, and he is not a draftsman until he does.

This idea of positiveness must be thoroughly absorbed by the student. Positive action must be a habit which controls his every move, which marks every dimension he prints, which directs every line he draws. Every line must mean something, must have a definite reason for existence, must be necessary to illustrate the idea which he wishes to convey to the workman, and every line must be a definite measurable distance from every other line, so that its location is fixed beyond a doubt. Lines which mean nothing, and cannot be measured, have no place on the drawing; they only confuse it.

A good picture of a machine could scarcely be called to the same service as a good drawing of it. The picture might give us an excellent idea of the machine, but for the purpose of the actual construction the picture is useless, while the drawing is of positive value. This value exists simply because of, and in proportion to, the completeness of detail which it shows. Hence in making a shop drawing the picture idea is entirely subordinate to the idea of *Utility*, the latter in fact being the measure of its value.

There are certain classes of drawings—of which the Patent Office drawing is a good example—in the making of which the picture idea is predominant. Here the purpose is to illustrate mechanisms, not construct them; hence the function of the drawing is in no wise that of the working shop drawing, and as such does not fall within our discussion.

Cost of Producing Drawings. The second general element involved in producing shop drawings is their cost, as measured by the draftsman's time. It is somewhat subordinate to the first element, for the drawing must be a good one, judged by an absolute standard, whatever the time or cost necessary to produce it. Cost, however, is an important item, and cannot well be overlooked. It is inevitable that in any enterprise economy will ultimately be sought, whatever extravagance an imperative original demand may have permitted. This is as true in the production of drawings as in the case of manufactured articles of trade. Drafting-room labor is a relatively high-priced service, and the salary list easily assumes considerable proportions, so that wasteful excesses count up rapidly. One of the qualifications of proficiency invariably required for this department of shop organization is rapidity of execution. This is not as dependent upon personal traits as at first might be supposed. A man may so husband his time and direct his efforts that he will easily distance his neighbor of more rapid motion. The latter may have less ability to make his energies count, and lack of judgment as to when just enough, and no more than enough, energy has been expended on his drawings. From the standpoint of *Utility*, the function of a drawing is fulfilled when it has reached the stage that it *completely* instructs; more time spent in elaboration is wasted, and is an unnecessary and therefore extravagant expenditure. The student must fully realize this. In his earnestness to produce finished and complete work he must constantly strive to accomplish results in the least possible time. This does not mean careless haste; far from it. A complete shop drawing cannot be made by short cuts, but through a systematic building of line on line, dimension on dimension. This is in sharp contrast to a haphazard habit of developing a drawing, first a line here and then a figure there, with no definite purpose in mind, and no hint as to when the drawing is actually completed.

The one method constitutes the efficient draftsman who works easily, receives a high salary, and is worth it, because he wastes no time in unnecessary labor. The other marks his unfortunate brother, plodding laboriously far behind, receiving a small pittance per hour, and worth less, because he does uncalled-for labor, and loses his definiteness of purpose in a maze of unexplainable lines and figures.

A working shop drawing, commercially considered, may well be defined as being "*Complete instruction from designer to workman issued at minimum expense.*"

This definition should be memorized by the student, and constantly kept in mind while making a drawing. The preceding pages should be re-read with this in view until the full spirit is appreciated.

The maxim as given above, if faithfully adhered to without modification, answers nearly every question that can be raised as to the excellence of a drawing. It can be used as a standard of judgment, whatever system of lines or symbols may be in vogue. It permits a draftsman to adjust himself to the rules of any shop or drawing-room, and yet produce a good drawing and satisfy his employer.

A drawing which is cheaply produced and at the same time does perfectly that for which it was made, that is, convey complete instruction, is beyond commercial criticism.

Method of Procedure. As the general objects to be attained in a working shop drawing have now been presented, it is necessary to indicate in detail how the work may be properly accomplished. In order to do this, it is proposed to produce systematically a full set of working drawings of a familiar and comparatively simple machine. The methods used will be those of a designing detail draftsman, producing commercial work fit for shop use. In the progress of the work, from its beginning in the rough, though accurate, pencil layout, to the completion of the tracings and the order sheets, the same bold style, clearness, directness and businesslike spirit which the shop atmosphere and surroundings would naturally supply, will be emphasized, and so far as possible imparted to the student. It is expected that the student will follow the text closely and study the plates carefully, endeavoring to familiarize himself with every detail illustrated. The more closely he is able to apply himself in this respect the more will he be able to partake of the life and spirit which is intended to be conveyed, and without which the true character of the work can be but poorly developed.

Incidentally, several purposes will be fulfilled by this treatment.

Ability to read drawings quickly and intelligently is almost as important as making them, and it is expected that the study of the plates, with a view to thoroughly understanding every line, will develop proficiency in the art of reading drawings.

The discussion in the text, of not only the form of the machine parts themselves, but also the tools and shop processes to produce them, affords considerable insight into the influences affecting good machine design. Without introducing any mathematical analysis or investigation, which is beyond the province of this book, much practical consideration as to the restrictions imposed by existing shop methods upon theoretical construction will be suggested, and the student encouraged to use his judgment thereon.

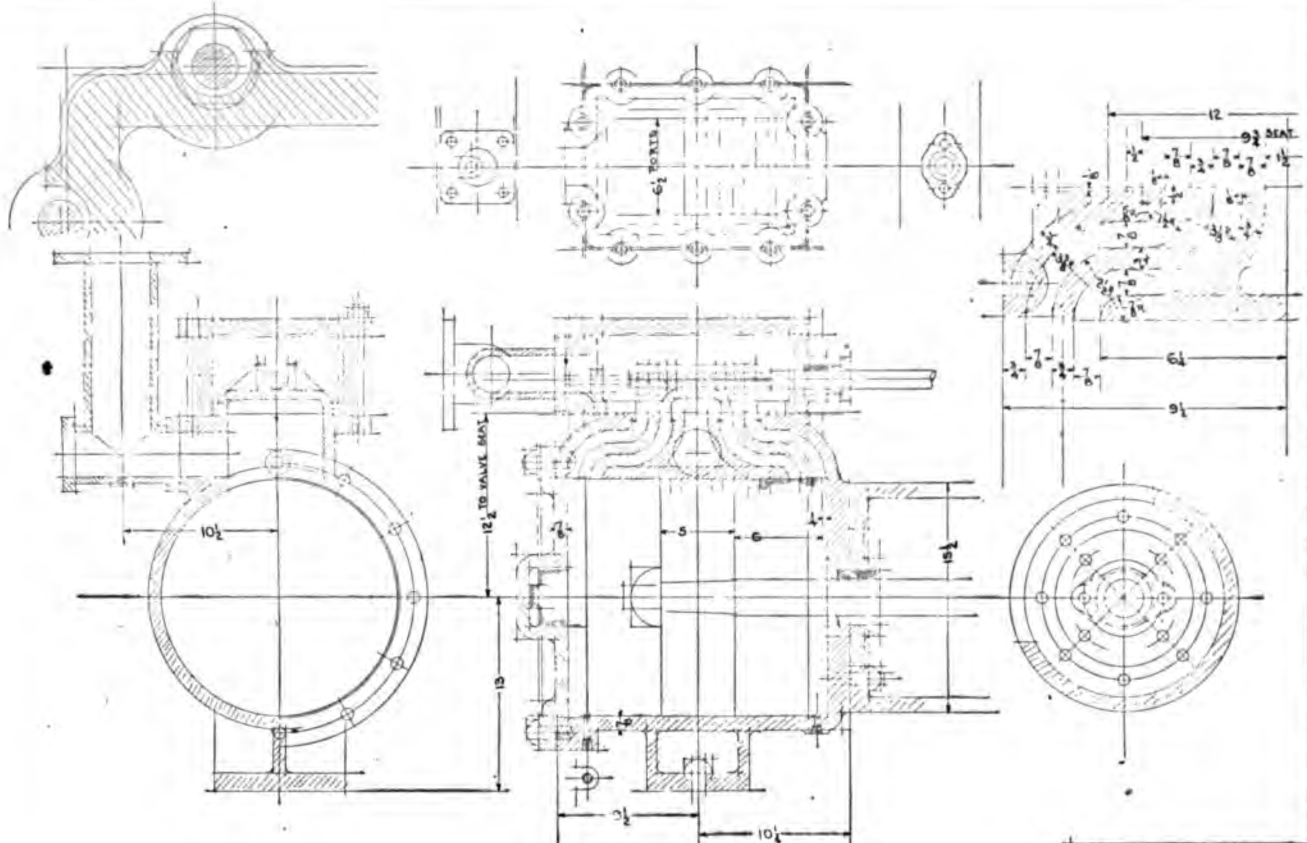
In the preliminary layouts the actual "sketchy" appearance of the pencil drawing will be imitated as far as possible, so that the student himself may imitate and catch the bold dash, yet fine accuracy, of the line-work, which is characteristic of the expert draftsman.

The completeness of a *set* of drawings is as important a lesson as the completeness of each drawing itself. In this is involved the proper arrangement and classification of details, the foundation layout, and the system of order sheets for getting work into and through the shops. This is a feature which very strongly affects some of the finishing touches to a drawing, for it is so easy to omit a "few last things" and turn in an uncompleted sheet. Every draftsman knows how many little things come up toward the close of a job involving complete drawings of a machine, and how strong the tendency is to omit them, and relieve himself of somewhat tedious details. The result is irritation and delay when the drawings get into the shop, and they return to the drawing room to be fixed up at a time probably inconvenient for all parties concerned. A good draftsman will turn in a *complete* set of *complete* drawings. It is highly important that the student grasp this idea, and study his work accordingly.

DUPLEX PUMP PLATES.

The typical set of plates chosen for this book in fulfillment of the above purposes, takes up the study of a simple, duplex steam pump. This particular type of machine represents the simplest and most elementary form of the steam engine in modern use in respect to valve gear and controlling devices. It is not an economical machine, yet its principles lie at the foundation of the economical high-speed engine, the latter being produced through a modification of the uneconomical valve gear such as is found on a pump of the type chosen, rather than through any radical change of construction as to the body of the machine. Hence the study of a steam pump may well precede that of higher forms of the steam engine. It is hoped that the study will so interest the student that he will be led to further investigation and development not only of the steam engine itself, but of that highly important division of modern engineering,—pumping machinery.

Thus we note another point of advantage in the study as out-



STEAM END LAYOUT
16·8½·12 DUPLEX PUMP
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE · 3"=1'

lined. The power end of the machine introduces us to the steam engine; the load end is the beginning of the engineering of pumping machinery.

Rating of Pump. A steam pump is rated by the bore of its cylinders and length of stroke, all being given in inches. A "$16 \times 8\frac{1}{2} \times 12$ pump" means that the steam cylinder is 16 inches in diameter, the water plunger $8\frac{1}{2}$ inches in diameter, and the nominal length of stroke 12 inches. These sizes are always given in the same order, beginning with the diameter of the smallest cylinder (in case there is more than one), then the diameter of water plunger, the common stroke of both being placed last. This expresses to the mechanic the rating of the pump in the clearest style and briefest language.

The pump illustrated here is designed for standard service, operating under a steam pressure not to exceed 100 pounds per square inch, water pressure not to exceed 150 pounds per square inch, and the rated capacity based on an average piston speed of 100 feet per minute being about 550 gallons. This requires that each side of the pump shall handle 275 gallons, and being double acting, shall make 100 reversals or 50 double strokes per minute.

Plate A. Steam End Layout. This plate illustrates, as nearly as reproduction can accomplish, the pencil layout of the steam end. It is the first work of the designing draftsman. The drawing as shown is exactly the type of layout which he would turn over to a detail draftsman, whose duty it would be to work up detail shop drawings therefrom.

The character of this drawing should be carefully studied. Remember that it is a layout, nothing more; also bear in mind that it is an exact, measurable working sketch. Attention is called to the sharpness of the lines, especially to the clean-cut intersections. Note the boldness, dash and businesslike style, the freehand cross-section lines roughly put in. There is no hesitation or worry as to where the end of a line shall be, or whether it crosses other lines which it theoretically should not. The intersections are allowed to indicate the termination of lines, and the rough section lines pick out the parts and separate them clearly to the eye. There is the spirit in this layout of confident, definite and rapid action, with no thought for absolute finish in line-work, but with every thought for absolute results as to measurable dimensions.

The data for the production of Plate A by the student are rather more complete than he would usually find in practice. Plates E, F and G show many details fully.

The steam cylinder and head, however, as shown in Plate G, are not dimensioned, and the student's problem is to produce this plate complete, with finish marks, dimensions and necessary data for a working drawing. In order to do this it is first necessary to work up Plate A with exactness, in pencil, and see that all parts go together properly. Then the detail of cylinder and head may be made separately by measurement of the layout drawing, and Plate G produced.

For this work the ordinary brown detail paper is very satisfactory. A hard lead pencil is necessary, as hard as 6H, and the point must be kept well sharpened.

There are two general rules of action in producing a drawing which give the answer to the question which oftenest confronts the beginner: "What is to be done first?" or "What is to be done next?" These rules are:

1. Draw everything that is positively known.
2. Work from the inside to the outside.

Every problem has some positive data, assumed or calculated, to start with. The first thing to do in every case is to get this data represented by lines on the paper. An expert designer has been heard to say that until he had spoiled the blankness of his sheet of paper by some lines, he could not design. There is something in this; and almost invariably the first line to draw is a horizontal center line somewhere near the middle of the sheet; draw it! Draw it at once without hesitation, and the layout is begun. We now have something about which to build.

In this case the designer would first calculate the size of the piston rod, and determine the fastening to the piston. He would then draw the rod and build a hub around it. He would next calculate the width or thickness of piston and size of packing rings, and draw the two vertical lines 5 inches apart, to indicate the piston faces. These lines would be limited by the cylinder bore, which he knows to be 16 inches; hence horizontal lines 16 inches apart, parallel to and symmetrical with the center line, are the next to be drawn. Short vertical lines indicate the location of the packing rings. As the nominal travel of the piston is to be 12 inches, the location of the piston and rings can be shown on both sides of the central vertical line at the limits of travel. A clearance must exist between the heads and the piston (in this case ¼ inch is allowed), hence the lines of the heads can be drawn, and the general inside outline of the cylinder barrel is complete.

This is all in direct application of the foregoing rules, and is so simple, natural and direct that it hardly requires such explicit statement. We have simply taken such data as we had and put it on paper, placing it where it can be seen from all sides, and where the mind is relieved of the labor of carrying it.

If the student will only appreciate this one rule and draw all he knows about the problem, he is well on his way to its solution. *Draw everything you know, and work for what you don't know*, is what these two rules say, and the first question to arise should be: "Have I drawn everything that is known about the problem?" before he asks himself or any one else: "What shall I do next?"

One other rule might be added to these two: *Keep dimensions in even figures, if possible.* This means that small fractions should be avoided. It is just as easy to bear this point in mind, and save the workman much annoyance and chance of error, as it is to disregard this matter. Even figures constitute one of the trade-marks of an expert draftsman. Of course a few small fractions, and sometimes decimals, will be necessary. Remember, however, that fractions must in every case be according to the common scale; that is, in sixteenths, thirty-seconds, sixty-fourths, etc.; never in thirds, fifths, sevenths, or such as do not occur on the common machinist's scale.

A systematic, definite mode of treatment on these lines must become a habit, so that all problems, however complicated, can be approached with confidence in the same way. It is the drawing of one line which makes clear the drawing of the next and subsequent lines; and the most serious obstacle which the student is likely to set for himself is trying to see the whole problem through from the beginning. Even an expert cannot do this, but allows the layout to develop results as he proceeds.

The details of the piston and rod being given in Plate E, the foregoing work is very easy for the student. The thickness of the barrel and heads being determined ($\frac{7}{8}$ inch in this case), the exterior outline may be partially drawn. The fixed head at the yoke end must be thicker than this, in order to receive the yoke and stuffing-box bolts without breaking through. The recesses or counterbores at either end of the cylinder should be so located that the packing rings run over the edge a little at the end of the stroke, thus preventing the wearing of a shoulder by the piston stopping in the same place every time. The counterbore should be deep enough to allow reboring the cylinder without the counterbore being touched by the tool. In this way the counterbore is retained to center the cylinder at its original location.

The size of steam ports having been calculated, they may be drawn in, the turns being made easy and as direct as possible. The height to valve seat must be kept at the lowest limit consistent with sufficient metal between and outside of the ports. As the detail of the ports might be somewhat troublesome, it is shown in an enlarged sketch for the student's benefit. Chipping or filing strips $\frac{1}{8}$ inch high are left on the port edges, which must be true, in order to finish them up easily.

The three inner ports are for exhaust, the outer ones for admission of steam. This five-ported cylinder is peculiar to the direct acting steam pump, it being a device to effect the cushioning of the piston at the end of the stroke, thus preventing the piston from striking the heads. This is necessary, since no positive limit of motion exists, as is the case in machines with crank and connecting rod.

When the edge of the piston has passed the outer edge of the exhaust port, as shown in Fig. 1, the steam, which has been exhausting through port A, is confined in space B and port C, and, being compressed by the piston, acts like a spring to retard its motion. If the point P is properly determined for a given speed, the piston will always compress the steam just enough to cause it to stop at the end of the nominal stroke; in this case, $\frac{1}{4}$ inch from the head. It is evident, however, that at different speeds the piston will have more or less power to compress the steam, and will not stop at the point desired. This causes the trouble of "short stroke," and consequent inability to make the pump work to its full capacity. Now if we connect ports A and C by a small opening shown dotted at D, and control this opening by a plug valve operated by hand from the outside, we can let a little steam leak by into port A, thus reducing the cushion and allowing full stroke.

In order to avoid complicating the drawing, no cushion valves are shown or required to be put on by the student. They are not customary in small pumps, but might advantageously be put on the present illustration.

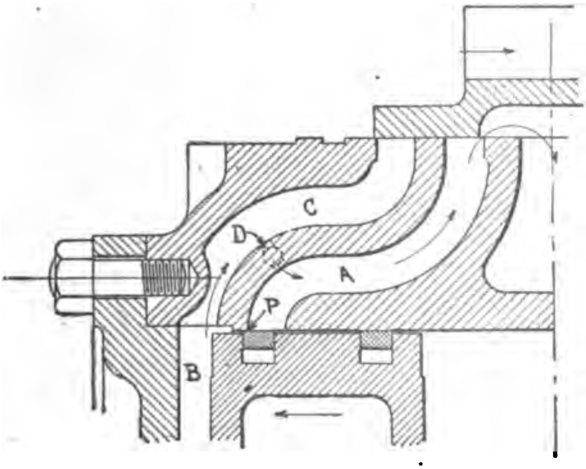


Fig. 1.

The valve seat must be a scraped surface, while the chest face need not be; hence the latter is finished $\frac{1}{8}$ inch lower. This also gives a ledge against which the steam chest fits, thus securing positive location.

The bolting of the heads and the steam chest should allow a width of packing inside of the bolts of $\frac{1}{2}$ to $\frac{5}{8}$ inch, otherwise there is danger of the steam blowing out the packing and causing leakage around the bolts. The bolts do not fill the holes, the latter being drilled large, from $\frac{1}{16}$ to $\frac{1}{8}$ inch. The spacing, if wider than 5 or 6 inches, is likely to permit springing of the flanges between the bolts, and consequent leakage. Bolts less than $\frac{5}{8}$ inch diameter are not desirable, as they can be easily twisted off with an ordinary wrench. In this case the cylinder head takes ⅞-inch bolts, the yoke, stuffing-box and gland, ¾-inch.

The flanges of heads and cylinders are usually from 25 per cent to 50 per cent thicker than the body of the casting.

Drips, ½-inch pipe tap, to be fitted with cocks, are necessary at both ends of the cylinder to readily drain the cylinder of water.

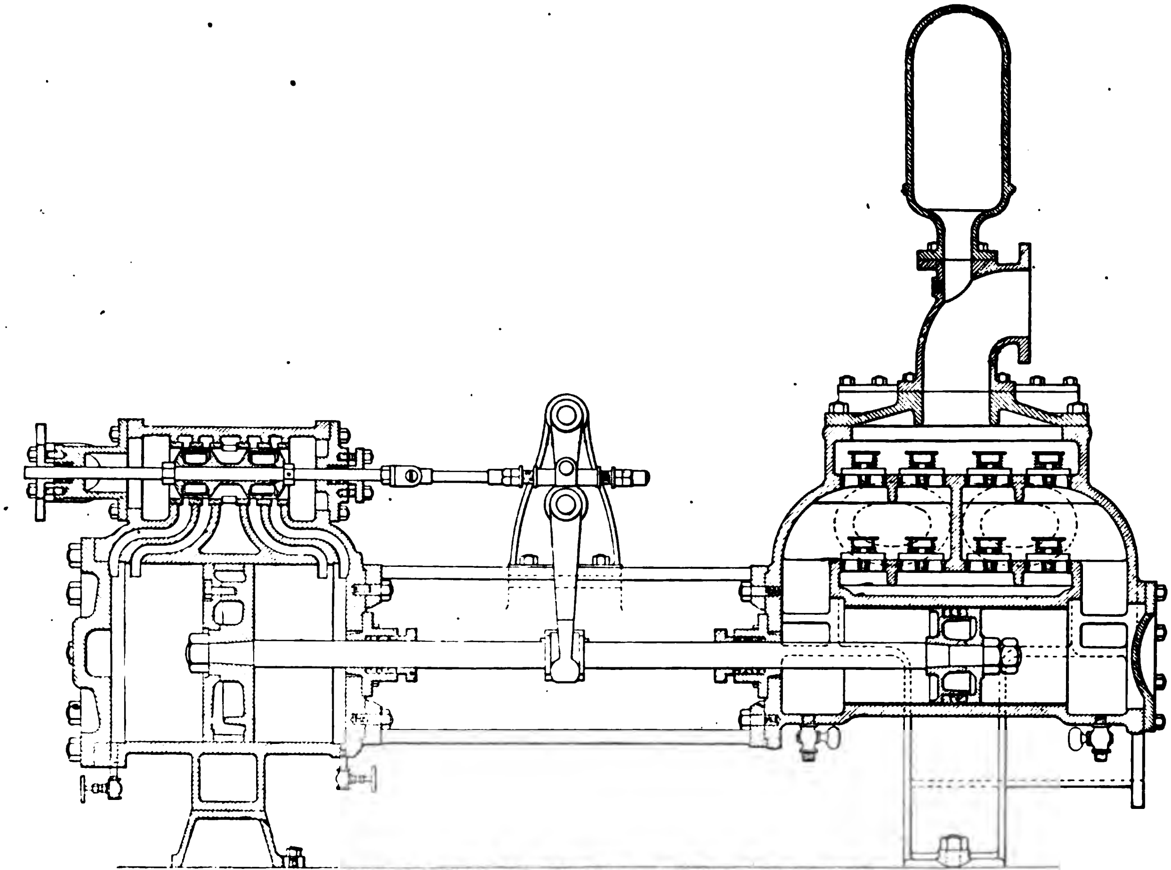
Molding of Steam Cylinder. The design is often influenced by the way in which the piece is to be cast. It often takes but a slight change of design to save many dollars in pattern making and foundry work. Hence the habit should be formed of always judging the design of a piece from the foundry standpoint. In this case it is evident that the ports and cylinder bore must be cored out, and the most obvious position of molding is to lay the cylinder on its side, the parting line of the flask being along a vertical plane running lengthwise through the middle of the cylinder. This permits the chest flanges to draw nicely, likewise the ribs on the foot, and allows the thin curving port cores to stand edgewise in the mold.

Another method of molding would be with the valve seat down. This would involve loose pieces for the chest flanges, and setting of cores for the cylinder foot. It would, however, assure sound metal beyond question at the valve seat. Spongy metal at the important wearing surfaces, the valve seat and cylinder bore, is not permissible in any case, and care in molding, and good design, is necessary for good results.

All corners must be carefully filleted, and chunks of metal must be avoided, especially where several walls or ribs join. The metal must be kept of average uniform thickness, so that the whole casting will cool uniformly.

Machining of Steam Cylinder. The boring may be done on a vertical boring mill, the heavy arm carrying the tool being thrust down unsupported into the cylinder, the latter being rotated by the table to which it is clamped. If the horizontal boring machine be used, the hole through the inside head for the stuffing box must be large enough to permit a stiff boring bar to be passed through. This allows a support at each end of the bar, to take the strain of the cut.

The plane surfaces may be finished on a reciprocating planer



CROSS SECTION OF DOUBLE ACTING STEAM PUMP.

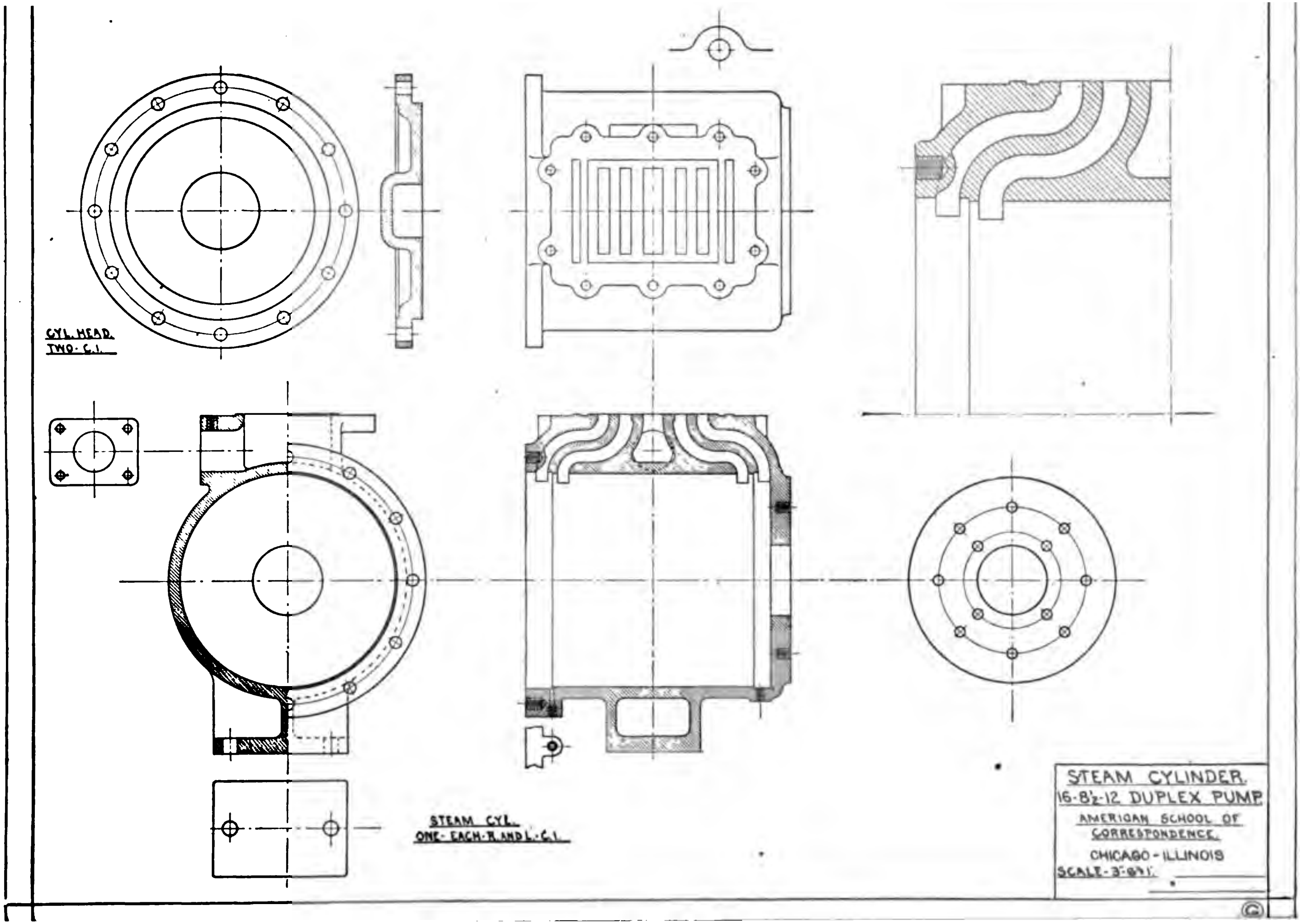
or a rotary planer. In the latter case it is desirable to keep all lugs or projections back from finished surfaces, to permit the large round head which carries the cutters to pass over them without interference.

The drilling of standard machine parts of this character is usually done through jigs, or plates carrying hardened steel bushings laid out to correspond with the holes required, and through which the drill is guided. These plates are located by some fixed line or lug on the casting, and then clamped fast, thus assuring exact duplication and rapid drilling, and avoiding the tedious laying out of the holes. In order to save changing the drill it is desirable, if possible, to maintain the same size of hole on any given surface. Of course it is not always admissible to do this.

Plate G. Steam Cylinder. After the exact and complete development of the steam-end layout, the student should be pretty thoroughly acquainted with the details of the cylinder. All the work thus far has been entirely for *his own* information, to get his ideas in visible shape, so that he himself can have a permanent record of them. This layout, however, is not in suitable form to finish up into a detail drawing. Its sketchy nature and the confusion of parts, especially if attempt were made to add dimensions, would render it somewhat difficult to be read by a workman taking it up as an unfamiliar subject. Hence it is now necessary to separately detail the parts, with the object in view of transferring, in the simplest and most direct manner, specific information to the workman which will enable him to construct the several parts. It is not enough now that the drawing be clear to the man who makes it; it must be clear, *absolutely clear*, to the shop mechanic, who has no means of knowing the designer's plans except through the information which the drawing gives on its face.

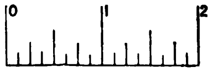
This requires that the draftsman should put himself in the workman's place, and forestall, by the explicit nature of his drawing, all possible questions which may arise in the shop. In this way only can he hope to avoid errors of construction and the continual annoyance of endless explanation of his orders.

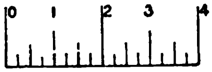
Plate G is to be a finished drawing, and the first thing to do is to lay out the sheet. The standard sheet for details which has

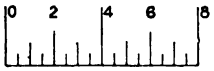


STEAM CYLINDER.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE-3"=1'.

been adopted is 18 × 24 inches trimming size, with $\frac{1}{2}$ inch margin all round, so that the working space is 17 × 23 inches. The rectangle for the title is to be laid off $2\frac{1}{2} \times 4$ inches in the lower right-hand corner, and must never be altered, either in size or position. This does not mean that other sizes are wrong, but once a standard system is adopted it must be strictly adhered to, both for artistic and commercial reasons. The scale to which the drawing is to be made is indicated in the title corner on every plate.

The scales permissible for shop drawings in the United States are those readily derived from the common foot rule, such as full size, 6 inches = 1 foot, 3 inches = 1 foot, $1\frac{1}{2}$ inches = 1 foot. These are the most common, most easily read from an ordinary scale, and one of these can usually be adopted. The student should learn to read these from an ordinary scale without being confined to a special graduation. To do this it is not necessary to divide each dimension by 2, 4 and 8 to get half size, quarter size, or eighth size, and then lay down the result. For half size, or 6 inches = 1 foot, $\frac{1}{2}$ inch on an ordinary rule represents 1 inch. Hence, each half inch may be read as 1 inch, and its subdivisions accordingly, thus: 

For 3 inches = 1 foot, or quarter size, $\frac{1}{4}$ inch represents 1 inch, and looks thus: 

For $1\frac{1}{2}$ inches = 1 foot, or eighth size, $\frac{1}{8}$ inch represents 1 inch, and looks thus: 

It is very easy to get accustomed to this, and it saves much time and trouble hunting up a special scale every time.

The other allowable scales, less common, but sometimes necessary on large work, are 1 inch = 1 foot, $\frac{3}{4}$ inch = 1 foot, $\frac{1}{2}$ inch = 1 foot, $\frac{3}{8}$ inch = 1 foot, $\frac{1}{4}$ inch = 1 foot, and $\frac{1}{8}$ inch = 1 foot. To use these scales conveniently, special graduation is desirable.

The general arrangement of the sheet, number of views and approximate space occupied, should be blocked out first. This can easily be done from the original layout. In general, several cross-sections are preferable to a single view, which involves many dotted lines. Dotted lines are very convenient for showing invisible parts of an object, but they are often abused, and the drawing of a complicated piece made indefinite and confused thereby. As already stated, a working shop drawing is solely to convey information to the workman at the least possible cost. A careful consideration of this will settle the question of the number of views necessary, their character, and the amount of dotted line work desirable.

Never let the drawing become the master; always be master of the drawing. Do not draw an extra view if no use can be seen for it. Do not put in dotted lines if the detail is completely shown without them. Full lines, or lines which show visible portions must, of course, be shown completely.

The nature of the pencil work on Plate G should be the same as on the original layout; viz., sharp, definite lines and positive intersections. Above all things learn the habit of accurate workmanship, for it will save many errors and a vast amount of time. The draftsman must check himself at every line he draws. Slight errors in scaling will often throw parts out of proper relation to each other, and interferences, which the drawing does not show, will become apparent only when the parts get into the machinist's hands.

It is dangerous practice to project across from one view to the other. It only takes a slight irregularity or spring in the T-square to vary the location of lines very perceptibly from where they should be, and once out of scale from this reason it is almost impossible to work a view with any certainty. Rather than project across from view to view, the principal lines, at least, should be scaled off on each view, and it will be found that in the end time will be saved and greater accuracy secured.

It is not economical of time to finish one view before beginning another. It is better to take some single detail of the drawing and develop it in all views, in order to study it from all sides. What is completed in one view may be found to be totally wrong when developed from another side, and the time spent on the first view will be wholly wasted. For example, in the present case the steam ports should be drawn in side elevation, end elevation and plan, and when thus completed the mind can leave them and in a similar fashion take up the study of the flanges, then the cylinder foot, and so on. Thus again the draftsman is master of his drawing, for he is continually making it tell him whether he is right or wrong. If, on the contrary, he allows himself to look at but one side at a time, and works from that standpoint alone, it may lead him into many difficulties from which he cannot readily extricate himself.

Do not be afraid to use the eraser. The draftsman who hesitates to draw until he is positive that no change will be necessary, is likely to spend the greater portion of his time in unprofitable dreams, for he is attempting the impossible. A drawing is a means, not an end; and, as has been already pointed out, it greatly assists the draftsman in clearing up many doubtful questions which the imagination alone cannot do.

A bold attack of a problem shows the quickest path to its solution, even if lines must be erased again and again. It is a sign of serious lack of ability to hesitate in the use of pencil and eraser.

Attention is called to the simple, straightforward character of Plate G. Notice the almost entire absence of dotted lines; the enlarged section through the ports, giving ample opportunity for dimensions without confusion; the use of a half end elevation and a half cross-section,—the one to make clear the flange and bolt layout; the other to show the exhaust opening, the small auxiliary views (drawn at convenient points) of the exhaust flange layout, the cylinder foot and the drip boss.

A steam cylinder is a fairly complicated casting; and it would be an easy matter, by the use of elaborate views, the dotting in of parts already completely shown, and careless line work, to rob this drawing almost entirely of its clearness and directness of illustration. *Just what is necessary* (for clearness' sake) *and no more* (for cheapness' sake), is the whole matter in a nutshell, and is what determines its shop and commercial value.

Dimensions and Letters. A good line drawing can be spoiled by poorly arranged dimensions and hasty lettering. The five principal points to be kept in mind to develop excellence in this respect are:

(1) **System.**
(2) **Accuracy.**
(3) **Clearness.**
(4) **Completeness.**
(5) **Character.**

System. The habit of system in placing figures and letters on a drawing is the one element which, to a large extent, controls all the others. If the systematic habit is established early, the other requirements will be fulfilled more easily. A haphazard method will, on the contrary, just as surely prevent the successful cultivation of the ability to figure a drawing. In fact, if the haphazard habit is continued it will itself, by the dissatisfaction which it causes, soon compel the draftsman to change his occupation.

In the first place, whatever part of a machine detail is to be dimensioned, that particular part should receive attention until it has been completely figured. Do not jump from one point to another, putting in a figure here and another there. Stick to one thing until it is done.

For example, take Plate F and the simple detail of the steam pipe. Suppose we start with one of the square flanges. The first question is: "Where is this flange located?" This is answered by the dimensions 5 inches and 21-inch centers, which refer the face of the flange to the center of the pipe and the flanges to each other. The next question is: "What are the three dimensions of the flange,— length, breadth and thickness?" This is readily answered as shown on the drawing. The next question is: "What further description is necessary to completely specify the shape of the flange?" This is answered by the radius of the corners, $\frac{3}{4}$ inch R. Next, "What drilling or special feature exists in the flange?" This is answered by $1\frac{1}{16}$-inch drill, $3\frac{1}{2}$-inch centers, and the letter *f* to denote that the face is to be finished.

The round flange of this pipe is approached and figured in the same way, except that the location of the face is preferably referred to the face of the square flange by the figure $8\frac{1}{4}$ inches, instead of to the center of the pipe, because the planer hand will more naturally use this figure.

These flanges are now to be connected by a pipe involving two sizes. The main pipe is 3 inches diameter inside, 4 inches outside, and $\frac{1}{2}$ inch thick, running into the two branches by fillets and radii, as figured. The two branches are really one pipe, 2½ inches inside, 3½ inches outside, ½ inch thick, and sweeping down into the square flanges by 4-inch radius.

This systematic method takes longer to explain than to actually execute, but it is typical of the train of thought which must be followed on all pieces, simple or complicated, in order to properly place dimensions.

In general, it may be stated that all parts of a piece must be referred either to each other, or to some common reference line, or to both. Each part so referred must then be figured as a piece by itself, and then its connections to the principal structure. Thus, figuring a machine detail involves three things:

(1) Relative location of its parts.
(2) Proportions of these parts.
(3) Proportions of connecting members.

As in the original design of a piece so in the figuring of it, the draftsman must as far as possible put himself in the place of the workman, judging the methods and processes of construction and available tools. This will largely influence the arrangement of the dimensions. Of course it implies considerable experience in shop work, which some students do not possess. He can begin none too early, however, to learn to look at his work from the shop standpoint, and surely make it some better on that account.

Pieces must not only be systematically dimensioned, but regularly specified and called for by suitable titles.

A title should specify at least three things:

(1) Name of piece.
(2) Number wanted for one machine.
(3) Material.

To these might be added a fourth; viz., pattern or piece number. The latter is not specified on the drawings under discussion, because systems of pattern and piece numbering are so varied that little would be gained by developing one for this special study.

These titles should always be put on in the same way, as the workmen become used to a certain system and are likely to misunderstand directions if a regular plan is not followed. A good way to arrange titles is suggested on the plates, although there are others which might be used.

Bolts are usually specified by diameter and length under the head, the length of thread being to some standard system in use by the shop, unless otherwise called for. Bolts are specified on the sheet containing the piece into which they are tapped. In the case of through bolts, tapped into neither piece, they are preferably called for in connection with the principal member.

Accuracy. Of course the dimensions on a drawing must be accurate. It is, however, a very easy matter to make errors. To insure accuracy a figure must *never* be put down carelessly, and a constant watch must be kept that scaled figures add up to over-all dimensions. It will not do to rely on scaling alone, as a very slight variation from exact scale may throw two dimensions out with each other. In spite of all the care that can be exercised errors will creep in, and a final thorough checking must be given a drawing before it is pronounced complete. A good rule to follow in checking up is to "assume everything wrong until it is proved to be right."

Clearness. As in the line drawing itself, there must be absolute clearness of instruction by the dimensions. Any doubt as to what a figure is, or what it means, rules out that figure as part of the drawing. If a piece is made wrong because doubt of this character is transmitted to the workman, the draftsman is always held responsible for the error.

Figures should, in all cases, be placed where they can be most clearly read. They should be bunched on a single view as far as possible, but not when greater clearness demands that another view be used. It hinders the reading of a drawing materially if the eye is forced to jump over large spaces of the sheet from view to view, to catch the several dimensions of a small detail. Usually it is easy to so group figures as to avoid this.

It is a good plan to keep dimensions off the body of the drawing, when it can be done so conveniently. It is not worth while, however, to go out of one's way to do this, as figures in the open spaces of a detail do not at all destroy its clearness.

Extended notes on a drawing to make it clear should not be required, but they should be used without hesitation if any doubt exists. An explicit note of instruction is the final resource for clearness when the art of drawing fails of its purpose, as it sometimes does.

Completeness. A detail is completely dimensioned when it shows *all* the figures necessary for the workman. Anything short of this is incompleteness. As modern shops hold the draftsman solely responsible for the design, the mechanic is not allowed to modify it by filling in any omitted dimensions. The only way to be sure that all the dimensions are on is to systematically go all round a piece inside and out, according to the method suggested under the paragraph on "System."

It is a good plan to always bear in mind that not only the machinist is to use the drawing, but also the pattern maker. For the benefit of the latter, special attention is desirable in figuring the cores. This saves him some addition and subtraction. In general, it has been found that less chance of error exists if mathematical work is not required of the shopman, all necessary data being furnished on the face of the drawing.

Character. By character in figures and letters is meant uniform style, height and slope, and a certain boldness peculiar to the work of the expert draftsman. The last is difficult for the novice to acquire. The student should not be discouraged because his efforts do not look like impressions from printers' type. Artistic excellence is the result of long experience, but is based on character. If the student can once get character into his work, the artistic feature will, with careful and constant practice, gradually develop. It is safe to say that there is no one element of a drawing which more positively stamps it as the work of an amateur than the character of the lettering, and every attention should be paid to getting out of the apprenticeship stage in this respect. Freehand lettering only is permitted in the drawings illustrated herewith. Ruled letters are seldom found on any working drawings, as the element of time involved is so great that few shops are willing to pay for it.

Uniform style requires that if capitals only are used in titles, they only must be used in notes and elsewhere on the drawing. If lower-case letters are used, they must be used in every part of the drawing. One style should not be mixed with another. The height of the letters should be limited by two horizontal lines, and though practice may render the upper line unnecessary, it takes but an instant to draw it, and uniform height is then assured. A good height for titles of details such as are illustrated is $\frac{5}{32}$ inch. The height once chosen should be adhered to throughout the whole set. A medium, not a hard, grade of pencil (3H) will give the hand greater freedom. A great temptation exists to omit titles from the pencil drawing, simply inking them on the tracing. This is false economy of time, for in the end it will be found that enough time will be saved by the certainty with which the tracing can be made to more than pay for the labor on the pencil drawing. Again, it permits the tracing, in regular shop practice, to be made by cheaper labor than that which produced the pencil drawing.

Uniform slope is most easily acquired by the use of guide lines put in at frequent intervals. A small wooden triangle can be made, giving the required angle. The angle of the letters shown on the plates is 9 degrees, or about 1 inch slope in 6 inches. The question as to whether letters should incline backwards, forwards, or stand vertical, does not enter this discussion. Character is not affected by the slope. The student may choose whatever comes most natural to him, but having chosen, the character of his work will be spoiled if he varies it. The most difficult of the three is the vertical style; hence most draftsmen incline their letters. The backward slope is used on the plates of this shop drawing paper, thus giving the student opportunity to compare with plates in the earlier books, and follow his preference.

The effect of change of style, height and slope is shown in Figs. 2, 3 and 4, respectively. Attention is called to Fig. 5, which is a sample title, in which these points are corrected.

Principal Titles. The principal title of a drawing should contain at least seven items:

(1) Name of principal details shown.
(2) Name of machine.
(3) Firm name and location.
(4) Scale of drawing.
(5) Date of completion.
(6) Draftsman's signature.
(7) Filing number.

To these are often added others, but for purposes of filing and reference the above at least *must* be put on. The filing number may or may not be put in the title frame, but it is really a part of it. It is often put in the margin below the title.

An arrangement of title should be established and then followed exactly, without variation either as to location on sheet or detail make-up. Abbreviated words are always permissible in titles, provided the meaning is clear. Special care must be taken in punctuation, however, as a title, whether abbreviated or not, has an unfinished appearance if the periods, commas and other necessary punctuation marks are not included.

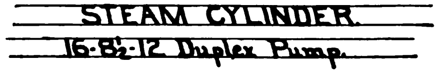

STEAM CYLINDER.
16-8½-12 Duplex Pump.

Fig. 2.

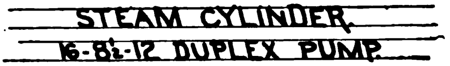

STEAM CYLINDER.
16-8½-12 DUPLEX PUMP.

Fig. 3.

The sample title illustrated in Fig. 5 indicates the arrangement chosen for the drawings of Part IV. Note that it is necessary in this special case to add an extra subject to the seven given above; viz., the residence of the student draftsman.

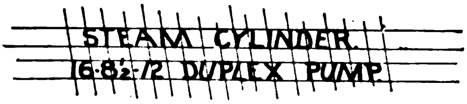

STEAM CYLINDER.
16-8½-12 DUPLEX PUMP.

Fig. 4.

This style of title must be put with care on every drawing, even on the rough pencil layouts. In the latter case it may of course be left in pencil, as the rough layouts are not to be inked.

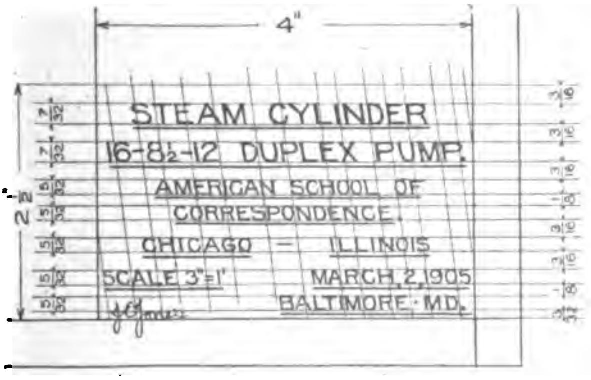


Fig. 5.

Inking and Tracing. Both bond paper and tracing cloth are used in business practice for finished drawings. It is desirable to keep a stock of both in any drawing office, so that either may be used as occasion requires. Bond paper stretched on the board gives a beautiful surface to take the ink, and very handsome and effective detail or assembled drawings can thus be produced.

Changes are not quite as readily made on bond paper as on tracing cloth, and it takes a little longer to make the blue print. In other ways the bond paper is not quite as flexible to use as the tracing cloth. However, one must be guided entirely by shop conditions to settle the question of preference. As the tracing cloth is generally used, and suits the purpose of the student better, it will be required in this work.

The inking should be done on the *rough* side of the cloth. One reason for choosing this side is that as the cloth tends to curl under toward the glazed side, the drawing as it lies right side up will tend to straighten itself. This seems to be a small point, but it is a very important advantage for filing and for the convenience of those who are to handle the drawings. Also the rough side takes colors and inks better than the glazed side. To trace on the glazed side is not wrong, for it is often done, but it possesses no advantages of its own, and has the disadvantage mentioned above.

Chalk dust scattered over the surface of the cloth after it is tacked down will remove the slightly greasy coating which prevents the ink from flowing well from the pen. This is always necessary if the glazed side be used, and usually for the rough side. The chalk must be carefully removed from the cloth before inking.

The first step in inking is to draw the center lines. Remember that *accurate intersections* are of the utmost importance. No circle is complete without two intersecting lines, preferably at 90 degrees, to determine its center, and these lines should be inked before the circle. When this is done a definite point exists for the needle point of the compasses. If the circle is drawn first the needle point may not be placed accurately at the center on the pencil drawing beneath, and the location be thrown out.

Likewise the principal center lines of pieces, the lines around which the pencil drawing was built up, should be at once put in.

The main body of the drawing, the full lines, should be taken next. In general, circles and arcs should be inked first, but there are cases where it is easier to run the arcs into the straight lines than to match the straight lines to the arcs. They are exceptions, however, and can be judged only as the case arises.

Straight lines, horizontal and vertical, should be inked with the T-square and triangle *in position*. It is a common practice to dispense with the use of the T-square entirely in inking in, using the triangle to match the lines to the arcs already drawn. A necessity for this implies very poor work on the arcs, for with any reasonable care true horizontal and vertical lines will match the arcs all right. With regard to time required, the accuracy with which the T-square may be brought up to a line, or the triangle set on the T-square, more than makes up for the time gained in even an approximate setting of the triangle without a guide. It is just as easy to cultivate the habit of holding the T-square and triangle with the left hand and the pen with the right, and draw an exact line, as to lapse into the other method, which is not workmanlike.

The lines of the body of the drawing depend for their width upon the size of the detail. For a large piece they may be $\frac{1}{32}$ inch wide, and the shade lines $\frac{3}{64}$ inch. For a small detail such widths would be too great. Remember that *contrast* is the principal aim, and to produce it is the only reason why we use different kinds of lines on a drawing. Hence the greatest care must be exercised to prevent body lines from becoming confused with center or dimension lines, and *vice versa*. Also thick lines are desirable for the production of a bold blue-print.

Shade lines are shown on Plates I, K and N only. They are put on according to principles already explained. They certainly improve the drawing from an artistic standpoint, and the student should know how to put them on when desired. Whether or not it is desirable to adopt them on all working drawings is not the purpose of this book to decide, or even discuss. Almost always drawings can be made perfectly clear without them, and are so made and satisfactorily used in probably the majority of shops. Some shops are willing to pay for the extra time necessary to put on shade lines; this, however, is purely their own investment.

Cross-section lines are usually drawn at an angle of 45 degrees with the horizontal, and on sections which are adjacent to each other the slope should be in different directions. If three or more sections come together the width between section lines can be so changed as to indicate clearly the different parts. An example of this is shown in Fig. 6.

The spacing of section lines must not be too fine, rarely closer than $\frac{1}{16}$ inch, more often from $\frac{3}{32}$ to $\frac{1}{8}$ inch, else the labor involved is too great and uniformity practically impossible. It is a waste of time to rule in section lines on the pencil drawing; they may be sketched in freehand, as shown on the original layout of the steam cylinder. Even spacing concerns the tracing alone, and the student should train his eye to regularity as he traces. The thickness of section lines may be intermediate between that of center lines and body lines of the drawing.

Inking Dimensions and Letters. Extension lines may be dotted, as explained in Part III, or they may be fine, full lines, the latter method being illustrated in the series of pump plates in this paper. Dimension lines are also often made fine, full lines. If these lines are made full they should be made as fine as it is possible to draw them and still have them firm, clear lines. The same width should be used as for center lines.

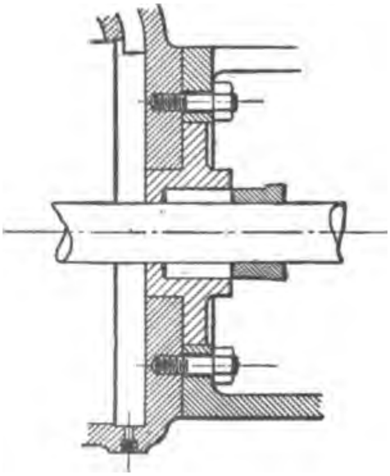


Fig. 6.

Character in inked figures and letters is more difficult to attain than in pencil work. In the first place a pen suitable to the style of drawing is necessary. A civil engineer's fine mapping pen, which gives character to his drawing, is not desirable in producing the bold character of a machine drawing. For the latter choose a rather stiff, blunt pen which is not "scratchy, but runs smoothly, making a line of uniform width. A pen with a round, or ball-shaped nib, recently put on the market, answers the purpose well for ordinary details. A bold, free stroke should be made with the idea of producing a smooth, even line, finished at the first trial. The hesitating uncertainty of the beginner's hand produces a "shaky" letter, and going over a letter or figure twice or more to smooth it up usually makes it worse.

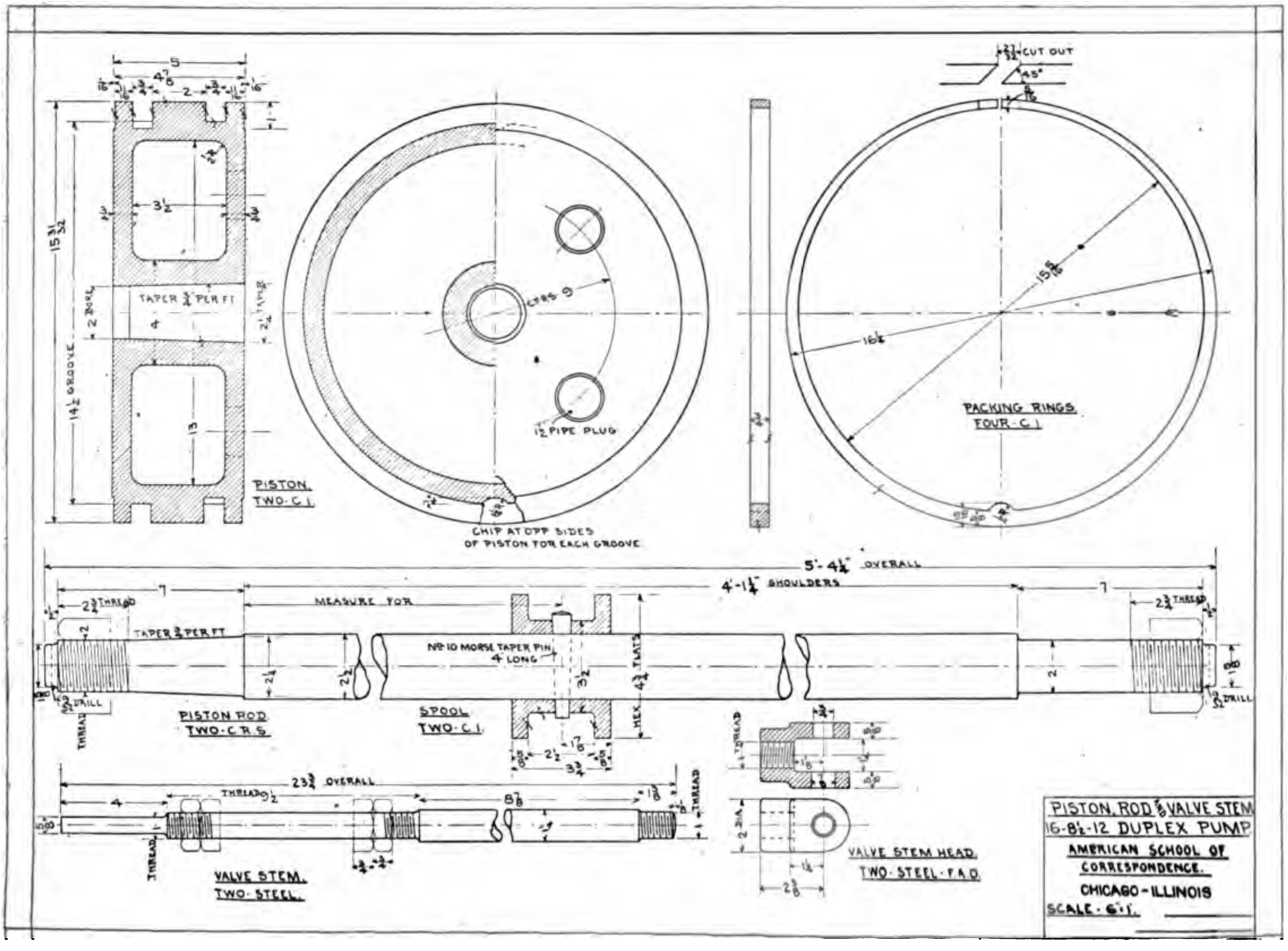
Figures and letters which are broad in proportion to height are easier to make, and have more character. It should never for a moment be forgotten that uniform *height* and *slope* carefully followed will develop character and quickly lead to artistic excellence.

Foot and inch marks are often put after figures according to the common usage. In cases where feet and inches are expressed, thus: 3′ — 6″, or 4′ — 0″, they are, of course, absolutely necessary, and the dash between the figures must be very positively indicated. In cases of inch dimensions alone the marks may be put on if desired, but where there can be no doubt that inches, and not feet, are meant, the inch marks are not necessary. This practice is followed on the plates of this paper.

Abbreviations. A list of the most common abbreviations in use on working drawings follows. This list has been adopted for the plates in Part IV:

F. A. O.	finished all over.
f.	finished surface.
R.	radius.
D.	diameter.
R. H.	right hand.
L. H.	left hand.
P. R.	piston rod.
P. TAP.	pipe tap.
CTRS.	centers.
C. I.	cast iron.
S. C.	steel casting.
Bz.	bronze.
C. R. S.	cold rolled steel.
T. S.	tool steel.
O. H. S.	open hearth steel.
W. I.	wrought iron.

Plate E. Piston Rod and Valve Stem. The piston is of the one-piece box type, with sprung-in rings. The width is reduced to $4\frac{7}{8}$ inches at the outside, so that if the piston strikes the cylinder heads it will not tend to spring and break off the narrow ridge of metal outside of the packing ring. The piston rod is fastened to the piston on a taper drawn in by a nut, and the



PISTON, ROD & VALVE STEM
16-8½-12 DUPLEX PUMP
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO-ILLINOIS
SCALE - 6"-1'.

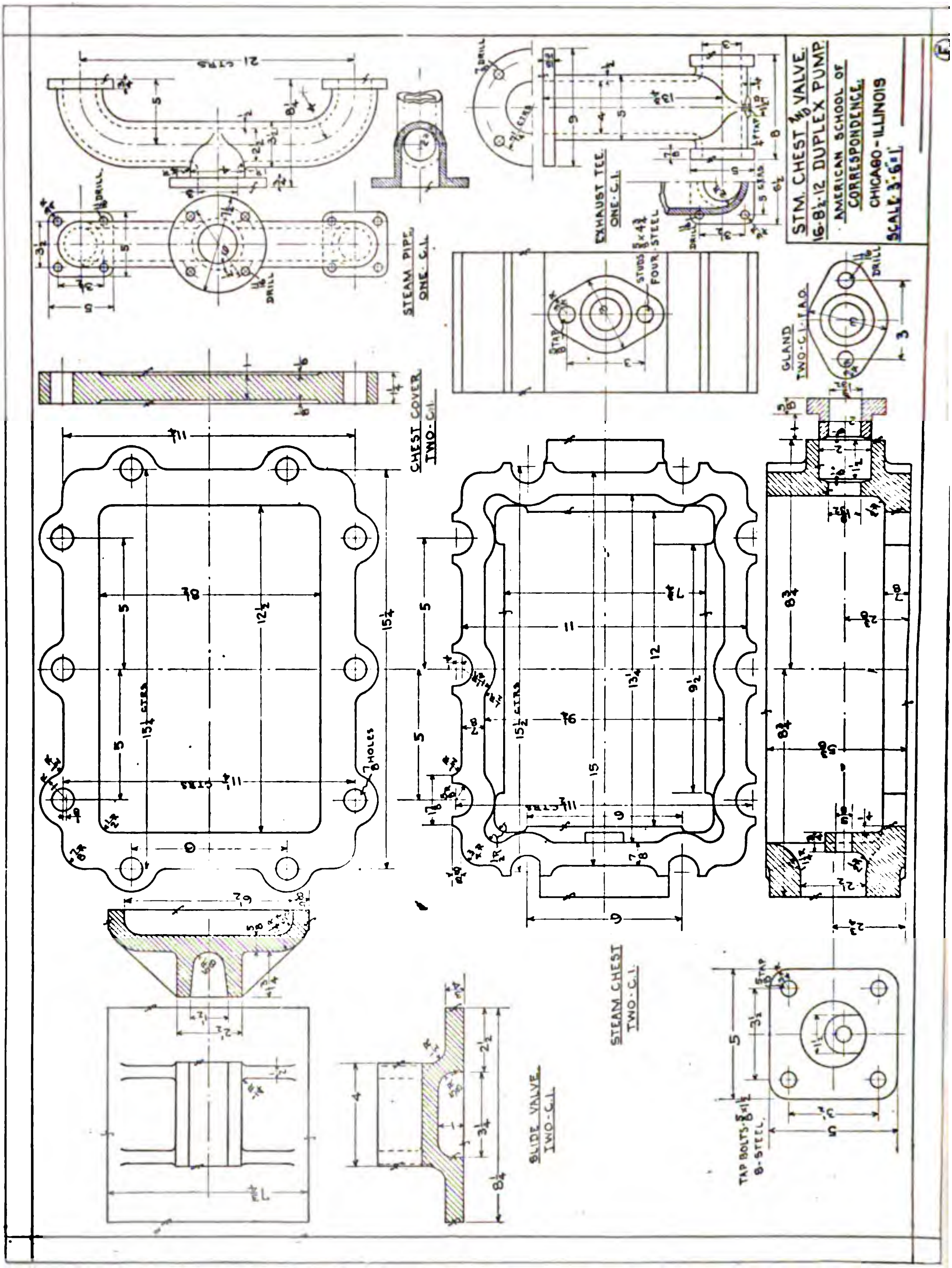
nut is checked by a ¼-inch split pin. The packing rings are prevented from slipping round the piston by lugs fitting loosely in chipped recesses in the groove. These being at opposite sides for each groove, the leakage of steam through the split in the ring is minimized, for it must pass half way round the piston before it can pass through the split in the other ring. This is a simple, but fairly effective, device.

The packing rings are usually cast in the form of a cylinder of some length, turned to a diameter a little larger than the cylinder bore, cut off to the required width, and sufficient space cut out to permit being sprung in to the size of cylinder bore.

The location of the spool on the piston rod is not positively known, as the setting of the valve bracket may be slightly different from what the drawing calls for. Hence, instead of a dimension, the words "measure for" are put on, to indicate that the spool be located during the erection of the pump. The hexagonal flanges of the spool are convenient to hold the rod from turning while screwing on the piston and plunger nuts.

Molding and Machining. There are no special features connected with the molding and machining of parts on Plate E. The holes in the piston side walls are necessary to give supports for the core, the piston being cast on its side. These holes, after the core is cleaned out through them, are plugged as indicated.

Plate F. Steam Chest and Valve. The steam chest in this instance is located on the cylinder by fitting down over the ledge made by the valve seat. The side flanges also serve the purpose of guiding the valve. It will be noticed that the steam chest cover is 15¼ inches × 11¼ inches, while the steam chest is 15 inches × 11 inches. This allows a ledge of ⅛ inch, all around which the cover overhangs the walls of the chest. The steam cylinder flange in order to correspond must likewise be 15¼ inches × 11¼ inches. The reason this is done is because of the difficulty of making good matched joints between the cylinder flange, chest and cover. The practice of thus leaving a little ledge all round is by no means universal, and often the irregularity in the joints is smoothed off by chipping. This is the case with the other flanges on this pump. The steam chest, however, was thought less likely to match properly, and the slight overhang gives the finished appearance of a sort of beaded edge.


STM. CHEST AND VALVE
16·8½·12 DUPLEX PUMP
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE 3"-6"=1'
F
STEAM PIPE ONE C.I.
EXHAUST TEE ONE C.I.
CHEST COVER TWO C.I.
GLAND TWO C.I. F.A.O.
STEAM CHEST TWO C.I.
SLIDE VALVE TWO C.I.
STUDS ⅝×4½ FOUR STEEL
TAP BOLTS ⅝×1½ 8-STEEL
7/8 HOLES
DRILL
CTRS

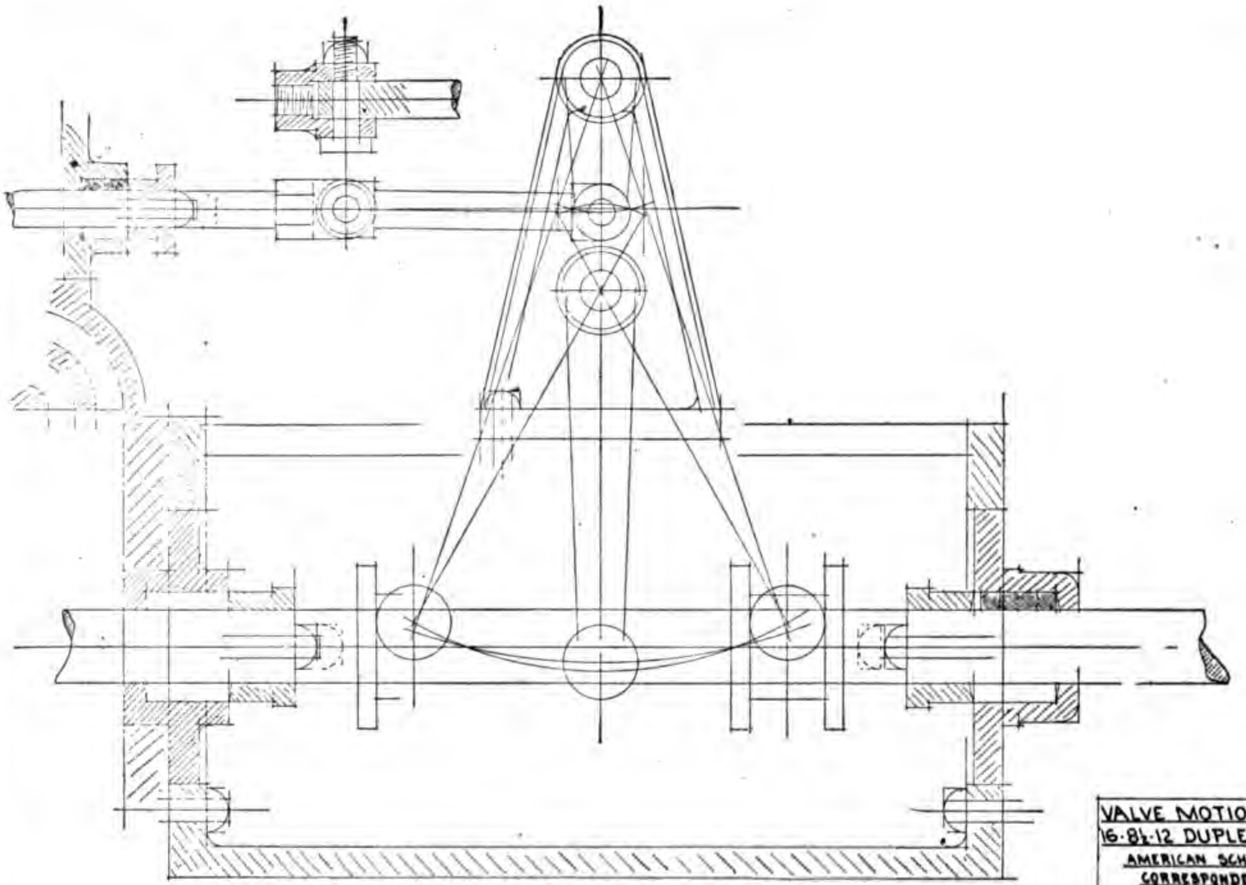
The valve is what is known as a "square" slide valve. This means that when the valve is placed central on the ports its working edges are "square" with the ports; that is, in exact line with them. If the valve be moved either way from this position, the slightest travel will admit steam to one end of the cylinder and exhaust it from the other. (See Plate A.) Another way of stating this is to say that a "square" slide valve is a slide valve without "lap."

The valve is driven from the valve stem by the striking of the nuts against the lug on its top. Since the valve is already guided on its edges by the steam-chest flange, the valve stem, to avoid springing, must be perfectly free in the slot cast for it, as is shown by the $\frac{5}{8}$-inch radius of the bottom, the stem being 1 inch in diameter.

The steam-pipe flange is made square to keep the height of the chest as low as possible. The radius of the bend should be ample; in this case 4 inches is considered sufficient.

The exhaust tee must have its upper flange high enough so that the chest cover can be lifted and slipped off the studs without interfering with it. The lower flanges should be made wide enough to permit the tap bolts to be put in without striking the 4-inch vertical pipe, 5-inch centers being necessary. The $\frac{1}{4}$-inch drip-cock, as located, readily drains the steam chest and exhaust passage of both cylinders, as well as the exhaust tee.

Molding. It is evident that the steam chest will be molded in the position shown on the drawing. The parting line of the mold will be through the centers of the steam-pipe opening and the stuffing-box. These holes must be cored out. The main body of the chest could be made to leave its own core, but it may not be made in this way. It may be cheaper to fashion the pattern solid, and make one large core-box for the inside. In this way the pattern will probably hold its shape better and require less repairs, than if it were made in green sand. The core-box will be an extra piece to make, but it probably will cost no more than to carve out the inside of the pattern, and is a rather more substantial job when done. The molding can be satisfactorily done by either method, shop conditions being the controlling element. As far as the labor of molding alone is concerned, the first method is probably easier, as it saves handling large cores.



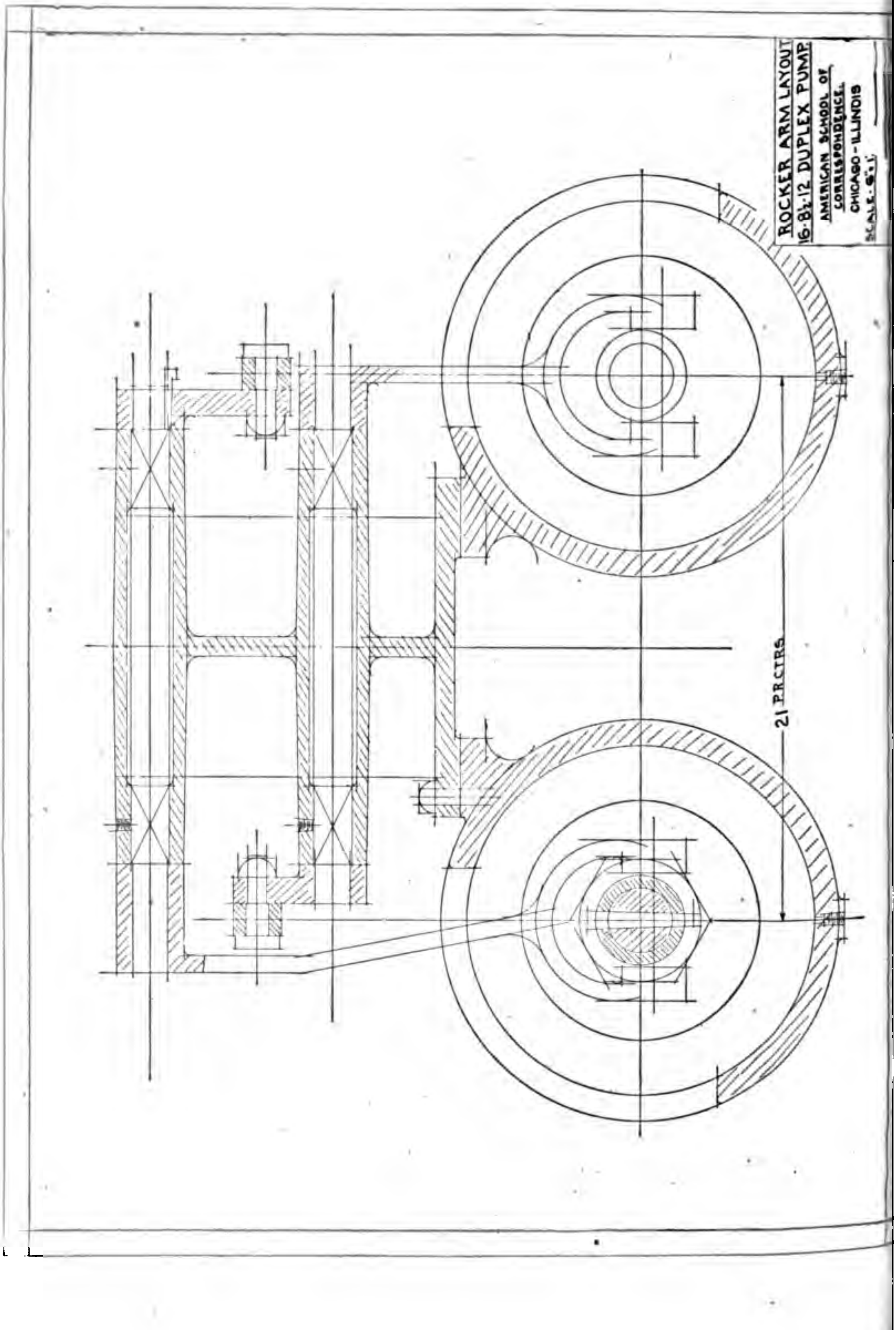
VALVE MOTION LAYOUT
16·8½·12 DUPLEX PUMP
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE - 6" = 1'

The other parts in Plate F are very simple in their molding, and require no special attention.

Machining. Most of the surface work on this plate is adapted to the planer. The slide valve may, perhaps, if finished in lots of considerable number, be more satisfactorily handled on the milling machine. The final finish of the face of the valve must be a scraped fit to its seat.

The drilling of the cover and pipe flanges is to actual layout on the casting, or preferably, through jig plates. A templet for laying out is at least desirable, even though the expense of a jig plate be not deemed necessary.

Plates C and D. Valve Motion Layout. These plates represent the layout of the valve motion, and are necessary in order to find the length of the levers and rocker arms. It will be noticed in Plate D that the valve stem of one side of the pump is controlled by the movement of the piston-rod of the other side, the proper direction of motion being given to the valve by placing the rocker shaft above or below the valve stem as required. By reference to Plate A it will be further noticed that the nuts on the valve stem inside the chest, which abut against the faces of the lug on the valve, do not rest against the faces of the lug in the position shown, but have considerable lost motion. This lost motion is one of the essential features of the valve motion of a duplex pump, and permits the valve to remain at rest for a short period at the end of the stroke, though the valve stem may have reversed its motion and begun its return stroke. When this lost motion is taken up by the movement of the stem and the nuts abut against the lug on the valve, the valve will move, and from this point to the end of the stroke be positively controlled by the motion of the stem. At the end of the stroke the stem will reverse, when the lost motion will again permit the valve to rest for the same period as at the other end, and then move on as before. The time of rest of the valve, and consequently the pistons and plungers, is approximately one-third the period of the stroke. This means that the piston on one side travels one-third of its stroke before it picks up, through the valve levers, the valve on the other side. During the second third of its travel it is bringing the

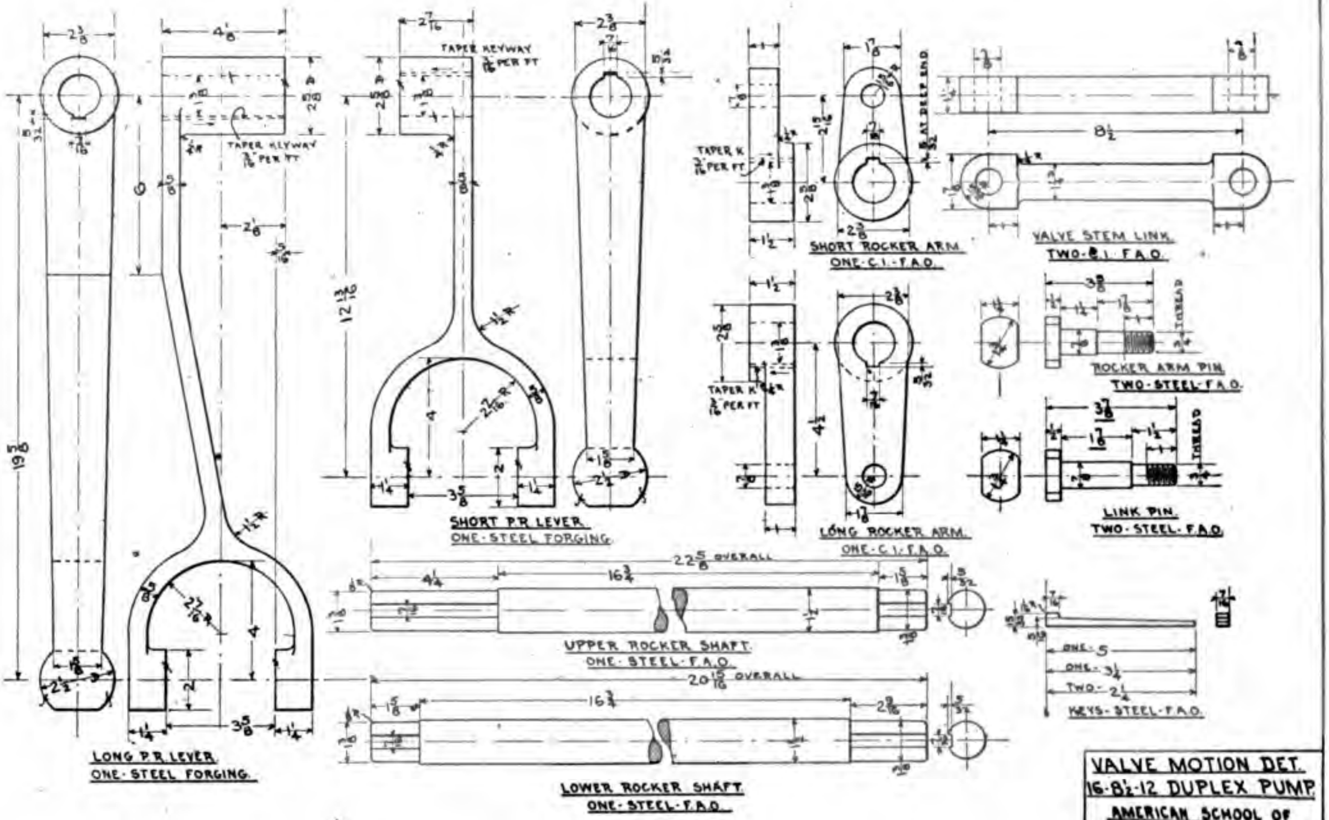

21 PR CTRS
ROCKER ARM LAYOUT
16·8½·12 DUPLEX PUMP
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO-ILLINOIS
SCALE: 6"=1'

valve to the point of opening. During the last third of its travel it is opening the port, wider and wider, to steam. Thus the opposite piston will start when the first piston has covered two-thirds of its stroke, and there will be only one-third of the stroke when both pistons are moving at the same time.

This relative period of rest to motion is not always made in this exact ratio, but is at least approximate to it. The period of rest at the end of the stroke is to allow the water end to adjust itself quietly to the reversal of motion about to take place at the end of the stroke. When the plunger stops, the water valves must be given time to seat themselves, and the flow of water through the passages checked. It is much easier to start the flow in the opposite direction if the reversal of plunger motion is not instantaneous. Hence for handling long columns of water, which, once in motion, tend by considerable energy to remain in motion, the duplex pump by this peculiar delayed action has been found to be well suited.

It will be found that for complete uncovering of port, and motion divisible into thirds as described, the travel of the valve stem should be three times the width of port, or $3 \times \frac{7}{8}$ inch $= 2\frac{5}{8}$ inches. A little more than this is allowed, and the travel made $2\frac{7}{8}$ inches in this case. Referring to Plate C, this distance is laid off as shown by the two limiting vertical lines across the line of the valve stem, the central vertical line of mid-position being drawn. The problem then is to find such centers for the rocker arms that the travel of the piston-rod spool will, through proper leverage, produce travel of the valve stem between these two vertical lines. This can readily be done by a few trials, the only requirement for this case being that the extremes of the arc of swing of both piston-rod lever and rocker arm shall be equally above and below the center of piston rod and valve stem respectively. The greatest possible travel of the piston-rod spool, $12\frac{1}{2}$ inches, is usually laid out in this case, not the nominal 12 inches.

From this layout the lengths of the levers and arms may be scaled off for the detail drawing, also the location of the rocker-arm centers. The student has the former given him on Plate K, but the latter, which is necessary for the development of Plate L, must be determined by his own layout. Plate D must also be



VALVE MOTION DET.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE · 6" = 1'

Ⓚ

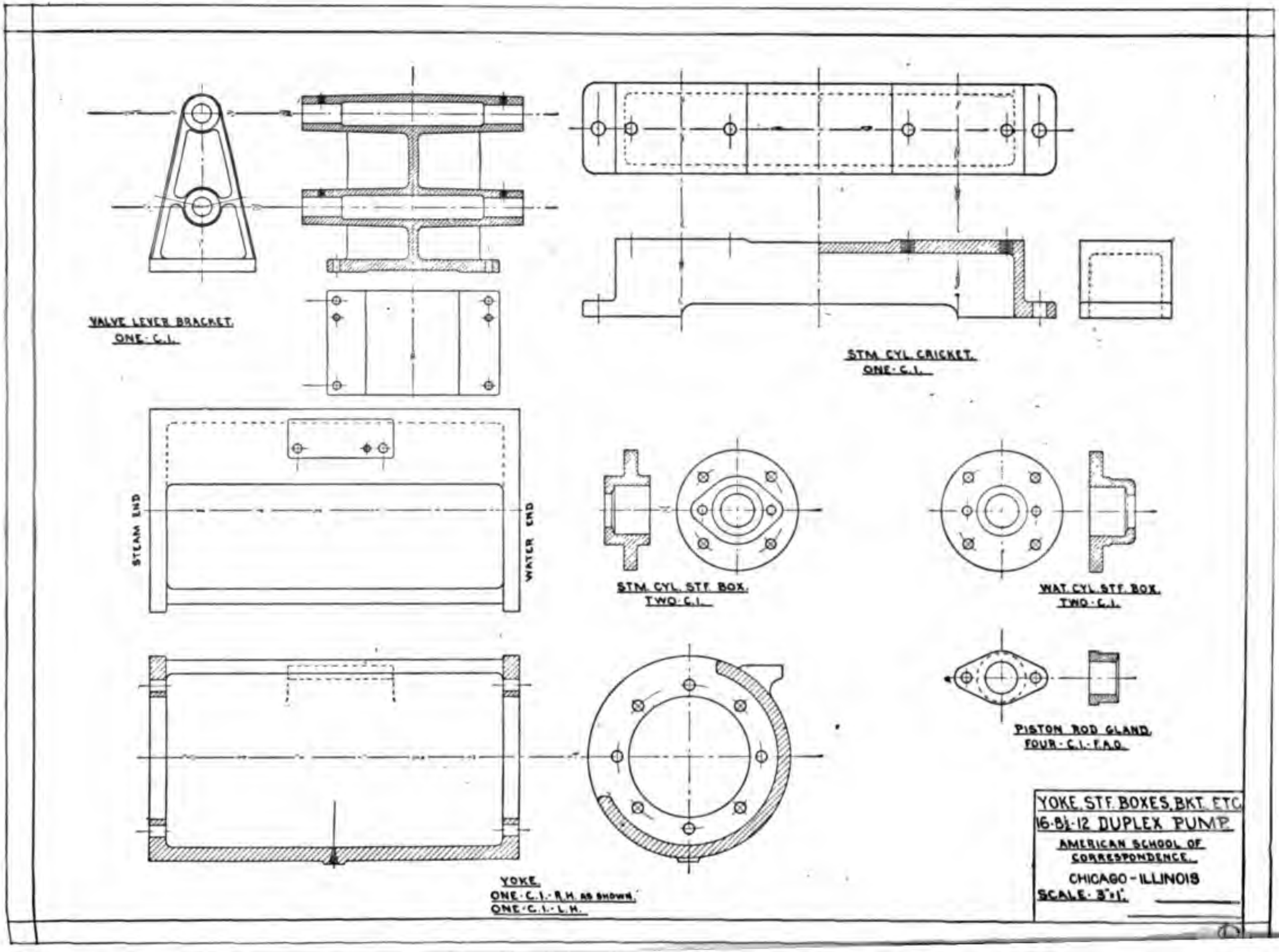
laid out before developing the cross-section of the valve bracket.

The design of stuffing boxes for both steam and water ends, and the length of the yoke, should be determined next. A safe method of assuming clearance between the spool and the gland studs at the end of the stroke, is to imagine that the gland stud nuts have accidentally worked off the studs, so that they are about to drop. They are thus shown by dotted lines on Plate C. A good clearance, say $\frac{1}{4}$ inch to $\frac{1}{2}$ inch, is then allowed, and the gland drawn in. The length of the gland is determined by the number of rings of packing necessary in the stuffing box; it is usually provided that the gland may compress the packing to about one-half its original depth before bringing up against the face of the box. Packing $\frac{5}{8}$ inch square will do for this size of piston rod, hence the faces of the yoke are easily determined, and its detail, with the stuffing boxes, proceeded with as on Plate L. The length of yoke may be brought to an even figure; and proceeding on the above plan the length can be conveniently made in even inches without any fractions; viz., 28 inches.

It will be noticed that the stuffing-box flanges serve to center the yoke in line with the steam and water cylinders. This is a desirable feature of construction, and forms a simple and easy method for lining up the steam and water ends.

Plate K. Valve Motion Details. The piston-rod levers on this plate are specified to be steel forgings. Forgings of this kind are expensive, but are light, neat and reliable for the important service which they have to perform. Castings, whether steel or iron, are much cheaper, and perhaps more commonly used for this detail. When sound they are equally serviceable, though of more clumsy proportions; but the danger in castings of this form is the existence of hidden flaws or pockets, which frequently occur at the points where the hub or the fork joins the arm. These flaws cannot be readily detected from the outside, and breakage may occur at some critical time, when the disability of the pump may be a serious matter.

The use of shade lines is illustrated on this plate. The increased artistic effect is noticeable, but it would seem that absolute clearness would still exist, even if shade lines were not used.



YOKE, STF. BOXES, BKT. ETC.
16·8½·12 DUPLEX PUMP.
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE·3"=1'.

It will be noticed that on the detail of the "link pin" two of the dimensions have a short "wavy" line beneath the figures. This is one of the several ways of indicating that the dimension is "out of scale." Some draftsmen use a straight dash beneath the figure; some draw a circle about it; some print after the figure, "out of scale." Although workmen are not allowed to scale drawings, but are required to "work to figures only," yet for general safety's sake, and for the sake of the draftsmen who consult the drawings frequently, attention must be called to any variation of the figure from the measured distance on the drawing. Nothing makes a workman, or any one else who reads a shop drawing, lose confidence in it more quickly than to discover that it does not "scale"; but when no indication exists that the draftsman himself is aware of it, then every dimension is viewed with doubt and hesitation, and the drawing becomes practically worthless.

Dimensions seldom should be out of scale; but if they are, through error or necessary change, a carefully worded note should be added.

Molding and Machining. No special features of molding or machining are noteworthy on Plate K.

Plate L. Yoke, Stuffing-boxes, Bracket, etc. Having worked up the layouts of Plates C and D, the student has enough information to proceed with Plate L. This, like Plate G, is without dimensions, the student's work being to make the drawing and fill in the necessary shop data.

The valve-lever bracket is bolted down to its lug on the yoke through holes larger than the bolt, thus permitting slight adjustment. When the proper location is determined, the bracket is positively fixed in position by two dowels, $\frac{1}{2}$ inch in diameter. The holes in both bracket and yoke are drilled through both pieces at the same operation. This very common method of fixing bolted parts of machinery in absolute position not only assures firmness, but also in case of removal, permits the part to be readily and positively replaced in its exact original position.

If possible, the steam cylinder cricket should be of such height that the stone or brick work upon which it rests shall be at the same level as that beneath the water cylinder. The tapped holes in the top surface receive bolts from the cylinder foot. These bolts are often used only for shipping purposes, the cylinder foot when the pump is set up being allowed to slide freely on the cricket, thus permitting free expansion and contraction. In such cases the water end is rigidly fastened to the foundation by holding down bolts.

Molding and Machining. The valve lever-bracket would most naturally be molded with the axes of the shafts vertical, the parting line of the mold being the center line of the middle web. This makes quite a long "draw" for the shaft bosses, but the ample taper on the outside overcomes this difficulty. The space between the side webs leaves its own core. The shaft cores stand on end in the mold, which is the best position for strength and stability.

Another method is to have the parting line of the mold on the vertical center line of the bracket, as shown in the end view. In this case the bracket would be cast on its side, and cores must be set for each side of the middle web. The shaft cores are set as easily as before, but in this case lie flat. As with the steam chest, each method has its advantages, which depend largely upon existing conditions. As cored work is generally avoided whenever possible, the first method would probably be chosen.

The shaft bosses are "chamber-cored," to save labor in boring, the bearing surface for the shaft being only a short distance at the ends. The chamber-core diameter should be enough larger than the shaft so that by no possibility can the cutter run into the rough scale, even if the hole be bored slightly out of line. If it should do this, the labor of caring for the cutters more than offsets the attempted saving of labor.

The yoke is simply a barrel open at each end, and with a piece cut out of its side. The inside evidently must be cored out, and the core is satisfactorily supported at the ends on its horizontal axis. The parting line of the mold may be either the vertical or horizontal axis of the end view, the only difference being that in one case the ledge for the valve bracket will "draw," and in the other case it must be loose on the pattern and "pulled in" after the main pattern is drawn.

The cricket and stuffing boxes present no difficulties. The bore of the stuffing boxes and glands should be from $\frac{1}{32}$ inch to $\frac{1}{16}$ inch larger than the rod, to allow the fit to be entirely between the rod and the packing.

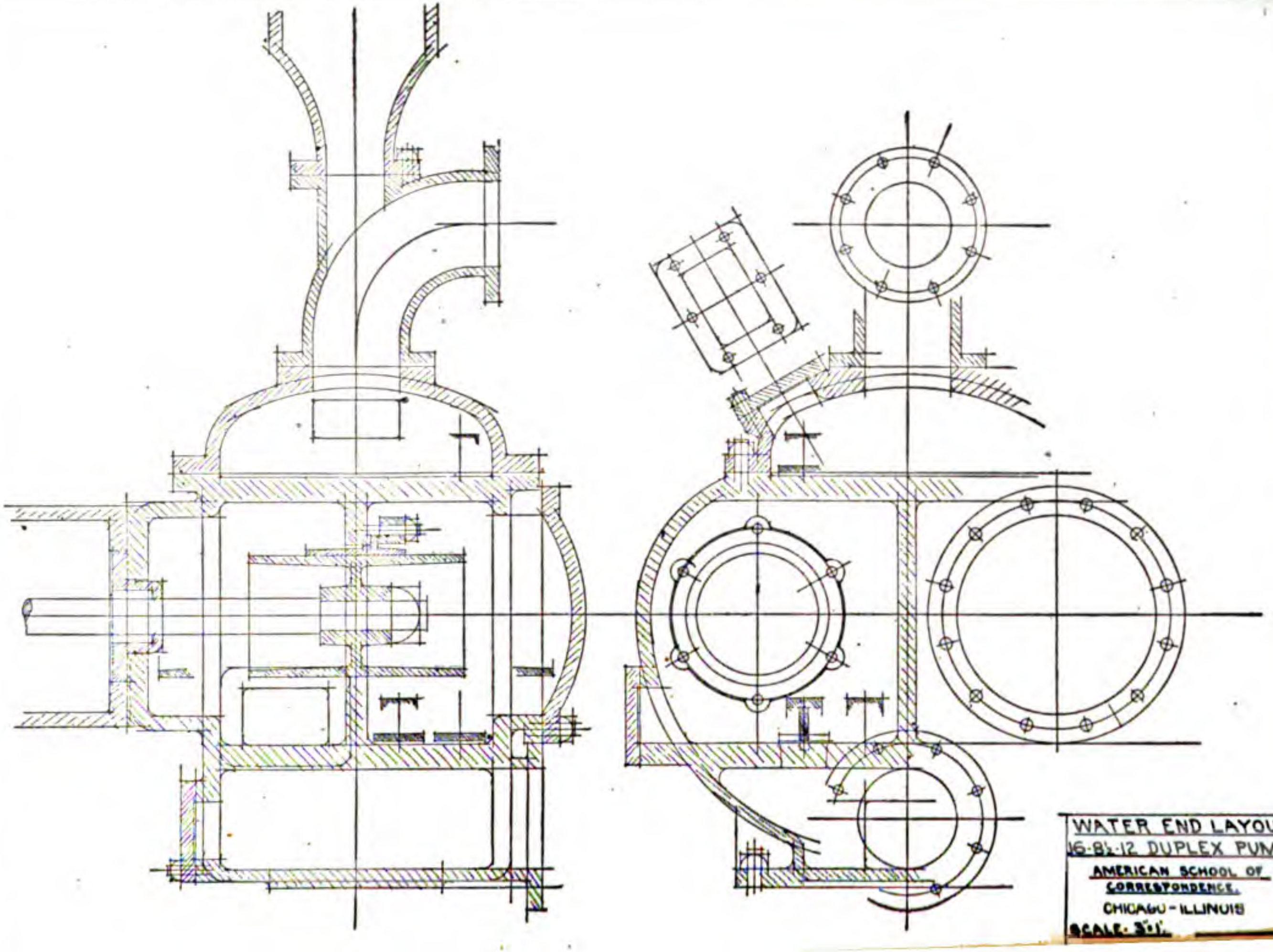
The horizontal boring machine with a double facing head is adapted to boring and facing the yoke flanges. The drilling is accomplished as before by templet or jig.

Attention is called to the tapped holes for oil or grease cups on the valve-lever bracket. The holes on the lower boss cannot be drilled strictly as shown, because the drill shank will not clear the upper boss. They should be swung around the boss at such an angle as will allow the drill to clear. This is a good instance of the common error of drawing details which cannot be made, and constant watch must be kept to avoid such mistakes.

Plate B. Water End Layout. As in the preceding work, Plates H and I being given in full detail offer a good start for the development of the water cylinder, which is the purpose of Plate B. As before, work should begin at the inside and progress outwards. Thus the piston rod with its nut should be drawn first, the hub of the plunger built around it, then the plunger barrel, the bushing, and ring to clamp the bushing. The limits of the plunger travel should be sketched in, and the valve outline shown in order to determine clearances. The progress of Plate B is on exactly the same basis as that stated in detail for the steam cylinder layout; hence it need not be repeated.

The points controlling the design of the water end must, however, be studied to enable the student to work intelligently. The fit of the rod into the plunger hub is loose, $\frac{1}{16}$-inch play being allowed, in order to permit the plunger to be guided solely by its bushing, and thus be independent of any change of alignment of the piston rod.

The relative length of plunger and bushing should allow the end of the plunger to overrun the edge of the bushing at the termination of the stroke, to prevent the formation of a shoulder. The bushing is made of brass because of the better bearing of the two dissimilar metals, brass and iron. Of course there is no lubrication except the water, and the dissimilar metals tend to "cut" less than if both were alike. The brass bushing also prevents the plunger from "rusting in" in case of long periods of



WATER END LAYOUT.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE- 3"=1'.

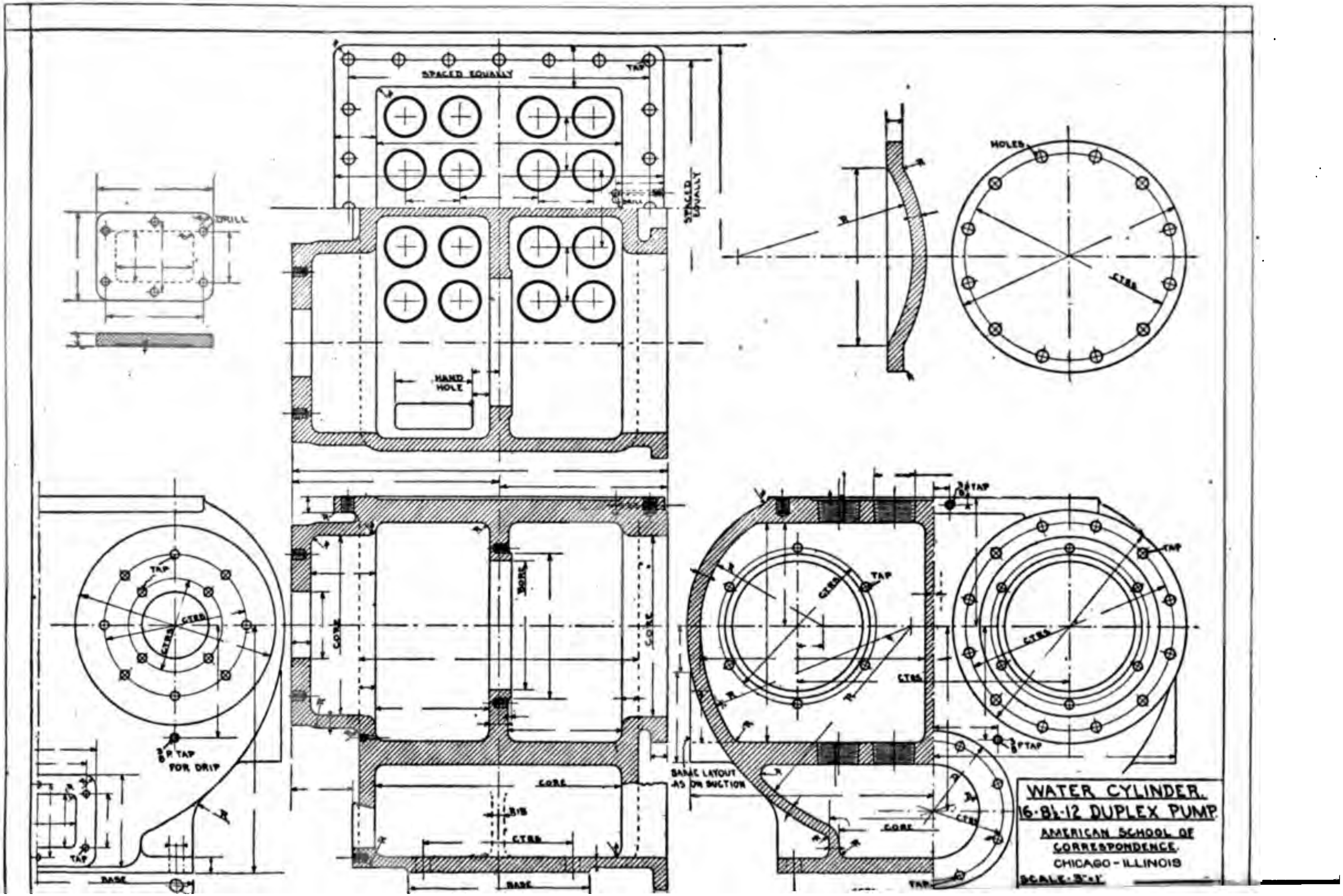
disuse. The bushing being of expensive material is made as light as possible, hence it has no stiffness of its own. Therefore it is reinforced by a deep cast-iron ring, which also takes the bolts and clamps the bushing tightly to its ground seat. These stud bolts are usually made of "tobin bronze," a rust-proof material, possessing strength almost as great as that of steel. This arrangement permits ready removal of the bushing when necessary.

As the parts of the common pump valve illustrated in detail on Plate H must be often replaced during service of the pump, provision must be made for unscrewing the stem and substituting a new one. This must be done through the hand holes provided on the cylinder. The lower valve deck must be located so that the inner valves when unscrewed will not strike the clamp ring. As shown in Plate B the clearance is pretty small, almost too small, but as it affects only two valves, it will probably cause no inconvenience. No hand holes are necessary for the end chambers, as access to the valves is had by removing the outer heads. The upper deck may be placed at a height giving sufficient clearance to allow the upper nuts of the clamp ring to be unscrewed with a socket wrench from the end of the pump. These decks are subjected to a severe pounding from the pulsations of the pump, and should be amply strong; 1¾ inches is deemed thick enough for this case.

The middle transverse wall may be 1½ inches thick and the middle longitudinal wall a little thinner, about 1¼ inches. With high pressures these walls, being flat surfaces and the valve decks likewise, are likely to fracture under the heavy pounding. To avoid making them excessively heavy they are often strongly ribbed, either on the inside or outside, usually the former.

The curving side walls are of better form to withstand pressure, and need not be as thick, 1 inch being sufficient. This can be decreased to ¾ inch in the suction passage below the deck, where little pressure exists.

The outer head is also considered strong enough at 1 inch thickness, on account of its curved shape. It requires ⅞-inch studs. Studs are preferred to tap bolts in this case, as in all other similar cases, on account of the frequent unscrewing of the nuts for purpose of removal. One or two unscrewings of a tap bolt in cast


SPACED EQUALLY
TAP
DRILL
SPACED EQUALLY
HAND HOLE
HOLES
CTRS
TAP
CTRS
CORE
BORE
CORE
CORE
RIB
CTRS
BASE
BASE
3/8" P TAP FOR DRIP
TAP
CTRS
TAP
CTRS
SAME LAYOUT AS ON SUCTION
3/4" TAP
3/8" P TAP
TAP
CORE
CTRS
WATER CYLINDER.
16-8½-12 DUPLEX PUMP
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE-3"=1'

iron will destroy the tightness of the thread, while the stud, being steel, stands the wear better.

The valve seats are taper screwed into the deck; they are sometimes forced in on a plain taper fit. They are located as closely as strength of the deck between the holes will permit. It is not well to place the edge of the valve closer than $\frac{1}{2}$ inch from the cylinder walls. The valve holes in the lower deck should be in line, or nearly so, with the holes in the upper deck, in order to allow the shank of the mill to pass through when milling the lower holes.

The suction opening is 7 inches in diameter, $12\frac{1}{2}$-inch flange, $10\frac{1}{2}$-inch bolt circle, $\frac{3}{4}$-inch tapped holes.

By means of the hand hole at the end of the suction passage, any dirt which may have been brought in through the suction pipe may be removed.

The water cylinder cap, discharge ell and air chamber may be laid out from the detail Plate I, and the student must do this to see that the parts actually go together properly.

With the foregoing discussion the student should be able to produce Plate B, which is the preliminary step to the detail drawing of the water cylinder as shown on Plate J.

Plate J. Water Cylinder. The water cylinder is, perhaps, the most complicated detail that the student will meet in this set of plates. Fundamentally, it is simply a box with curved sides, divided by the several walls into five compartments, each of which communicates with the outside by a round nozzle or flange. If this basic idea be kept constantly in mind, the student will have no trouble in building up the detailed design.

This fundamental conception of a complicated piece is a very important idea, and should be developed carefully by the student. It is one of the great secrets of good design, both from an artistic and a commercial standpoint. We often see a machine which seems to begin anywhere and end nowhere; it appears to be a miscellaneous collection of bosses, lugs, ribs and flanges. There is no general prevailing shape to the structure, no harmony of the lines. This is because the designer, if he may be so called, did not have the fundamental notion of shape, to which all minor details should have been subordinated. He simply grouped parts together, without considering the fundamental structure.

In this water cylinder the box is the basic part of the structure, and its lines must be first developed; they should be designed to convey a smooth, regular and consistent surface to the eye. Then the nozzles and flanges may be added as subordinate parts; they will merely interrupt, but not destroy, the prevailing outline of the box. The dotted lines in the cross-section views of Plate J show the general shape behind and beneath the nozzles.

The hand holes are the same as on Plate I, and the detail of the cover should specify the number required for both places.

Provision for draining the four chambers of the water cylinder is made by the $\frac{3}{8}$-inch pipe tap holes at the lower deck, and the cap, likewise, by the single hole at the upper deck. Drip cocks are screwed into these holes.

The holding-down bolts should not be less than 1 inch diameter; $1\frac{1}{4}$ inch would perhaps be better; and the holes in the foot should be drilled at least $\frac{1}{8}$ inch large.

Dimensions. It will be noticed that this plate has dimension lines, but no figures. This is because the cylinder is rather difficult to figure, and it is desired to guide the student in arrangement of the figures without lessening the benefit of his study of them. Special attention should be paid to this feature of the plate. Notice that although space for dimensions is restricted, a clear opening is always found for the figures; and when one view seems to offer no space for a figure, another view gives the desired opportunity.

No finish marks or titles are shown on this plate, these being left entirely to the student for insertion.

Molding. The centers of the curves for the sides being on the main horizontal axis of the nozzles, the cylinder, if molded to be cast vertically as shown, will draw readily both ways from this line. The exceptions to this easy draw are the foot, suction nozzle and flange, and hand-hole boss. On account of the inside of the cylinder being cored, these pieces if made loose on the pattern have ample space to be "pulled in" after the main pattern is withdrawn.

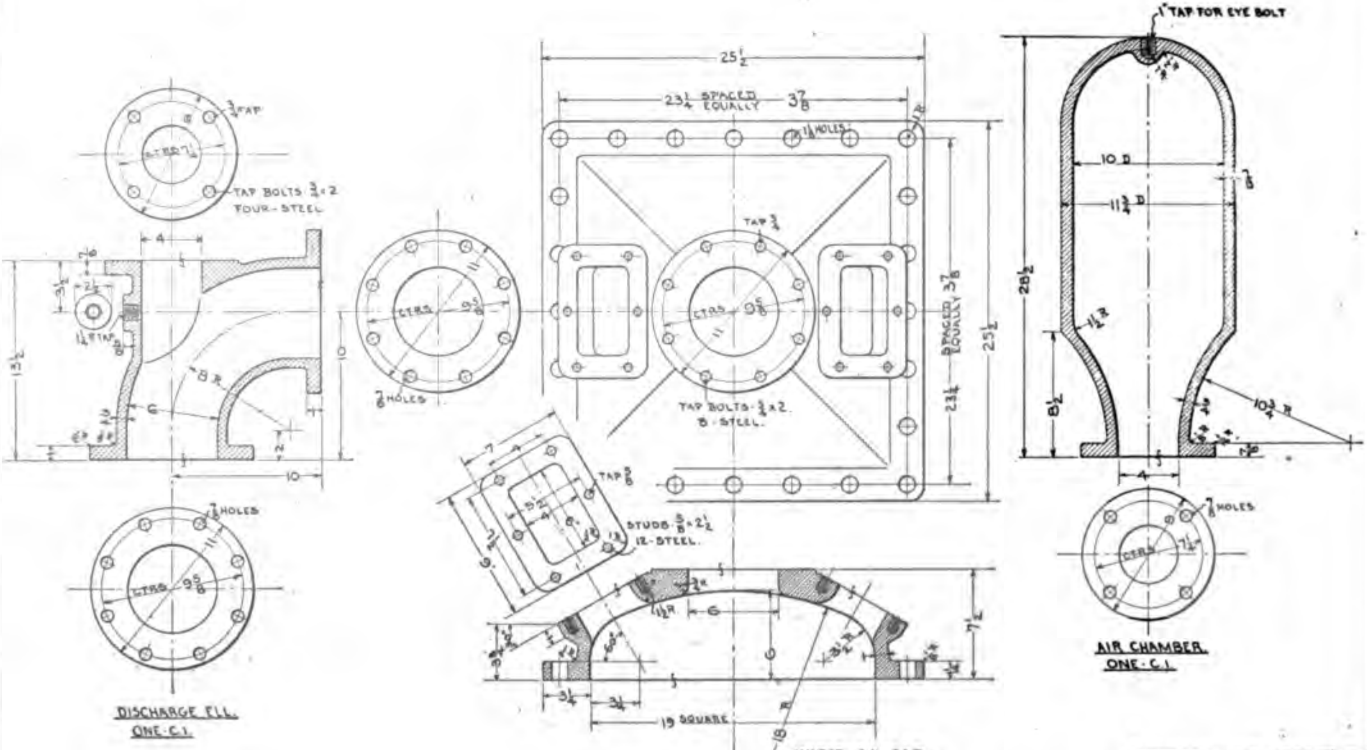
The suction passage below the deck communicates with the main core through the valve holes, hence it may be supported from the main core. This involves some difficulty, however. If a three-part flask be used, and another parting established at the center of the suction flange, in addition to the previous one, the problem becomes much simplified.

It is desirable to make the four chambers of the cylinder alike in general proportions. It is then possible to make a single core-box, and by the use of loose pieces change the length of the nozzle cores and transpose from right to left, thus saving labor on the pattern. This, however, multiplies the loose pieces on the pattern. The many pieces are likely to become lost and make frequent repair necessary. Hence it is not always wise to use a single core box too much, and good judgment is required to fix the limit.

Machining. Special double horizontal boring machines are now in common use for such cases as this water cylinder. The centers are made adjustable, so that within limits any distance between piston-rod centers can be met. The advantages of double boring are, of course, most obvious for a considerable number of duplicate cylinders.

It will be noticed that the face of the suction flange is carried out flush with the cylinder head face. This affords opportunity for finishing all the end surfaces at a single setting of the tool, whether the work be done on the rotary or reciprocating planer. This same point might have been observed on the small hand-hole boss at the other end of the cylinder, but the advantage gained did not seem to warrant extending the " reach " through the hand hole.

Plate I. Water Cylinder, Cap and Air Chamber. For a water cylinder cap of this size, the most difficult problem is to find room for the hand-hole bosses. A hand hole 4 inches × 6 inches is about as small as can be used, and this calls for a flange at least 7 inches × 9 inches. These are the proportions shown on the plate, and since the boss overhangs the bolts in the main-cap flange, it must be cut away underneath to clear the nuts. If three stud bolts are used on each side, this overhang also requires that the nut be " fed on "; that is, screwed on little by little as the end of the stud protrudes above the flange when the cap is being lowered into place. This is an awkward process, but is sometimes necessary.



WAT. CYL. CAP & AIR CHAMB.
16-8½-12 DUPLEX PUMP

AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO - ILLINOIS

SCALE-3"-1'

The discharge ell should have an easy bend; usually the radius is somewhat more than the outside diameter of the pipe, in this case 50 per cent greater. It is customary on this piece to provide an opening for the attachment of a relief valve as shown, 1¼-inch pipe tap. This valve can be set to open at a desired pressure, so that the water end may be relieved in case of accidental excessive pressure.

The air chamber provides an air cushion for the water to make the delivery more constant, and take the shock which would otherwise come with hammer-like force and full intensity upon the cylinder. Being placed at the highest point of the water end, air will naturally tend to collect in the air chamber and keep it charged. In some cases, however, a special charging device is necessary.

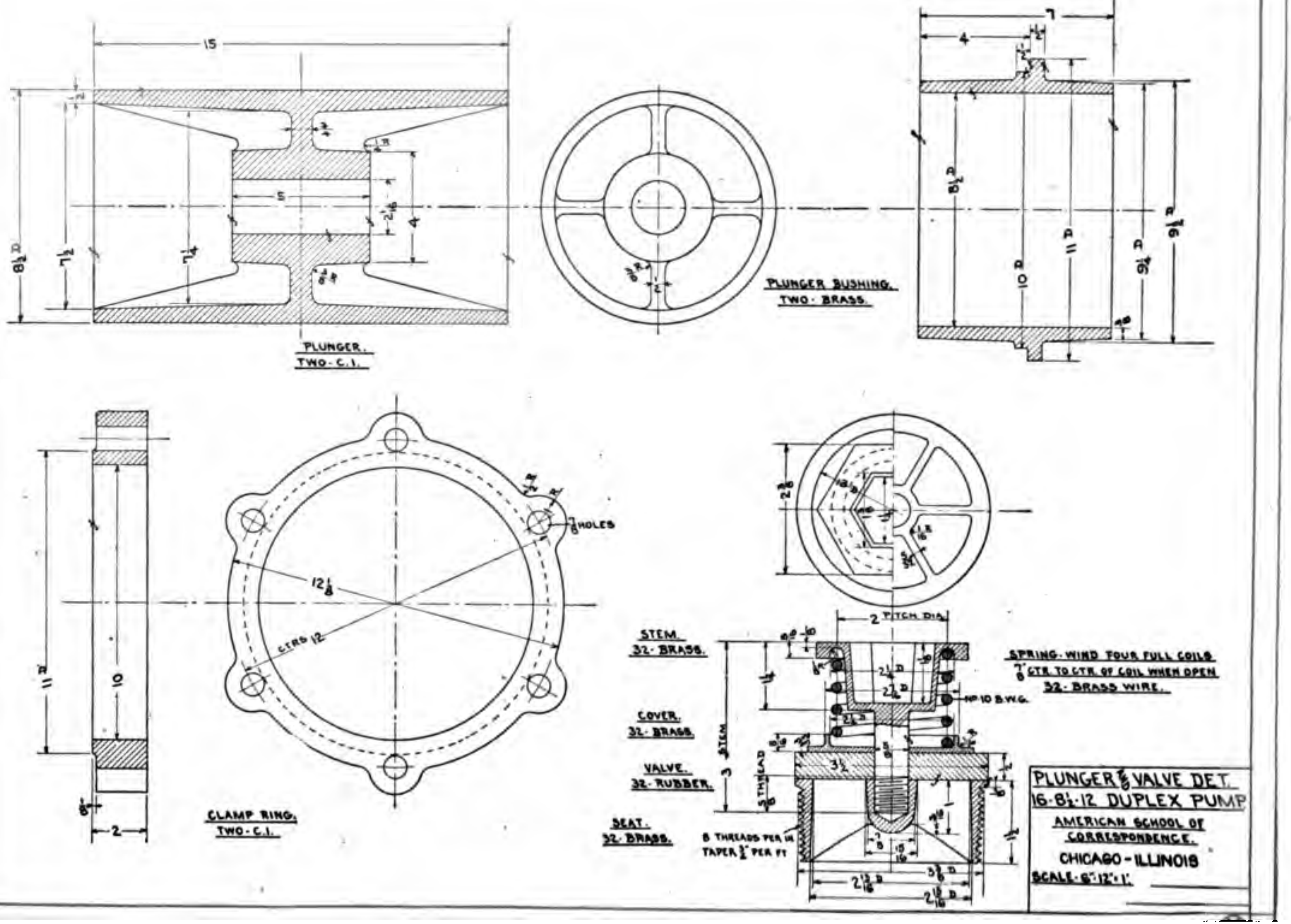
Molding and Machining. The hand holes being at an angle will not "draw." Hence cores must be set for these openings at least, and it may be desirable to core out the whole inside of the cap for the sake of keeping the pattern in good shape by making it solid. Otherwise it is easy to let it leave its own core.

The overhang of the hand-hole bosses requires loose pieces for the overhanging part. They are "pulled" in after the pattern is drawn.

The molding and machining which are further required on details of Plate I are simple, and require no special discussion.

Plate H. Plunger and Valve Details. This plate is noticeable for illustrating a method of drawing details not used elsewhere in this set of plates. On the other plates each piece is separately detailed. On Plate H the details of the valve, cover, seat, stem and spring are shown assembled, and dimensioned without separation. This is an allowable method when clearness is not sacrificed, but it is usually found desirable only with simple construction. It concentrates parts on the drawing, and probably saves some time, besides showing the workman just how the parts go together. The only test which the student need to apply in this, as in any method of detailing, is the test for absolute clearness.

It is believed in the case of the valve as shown that the details are completely illustrated without sacrificing clearness. Special care in putting in dimensions is of necessity required.



PLUNGER & VALVE DET.
16-$8\frac{1}{2}$-12 DUPLEX PUMP
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO-ILLINOIS
SCALE-6"-12"=1'.

The valve stem can be unscrewed either with a socket wrench on the inside or an ordinary fork wrench on the outside.

The seat, after being screwed to position in the deck, is often faced off, to true up any distortion caused by screwing in.

The valve itself, of rubber, can be bought of any desired grade of hardness. The specification for any given set of valves depend upon the quality of the water, the pressure and the general service of the pump.

Molding and Machining. By reason of the simple nature of the parts on this plate, the molding and machining is left entirely to the original consideration of the student.

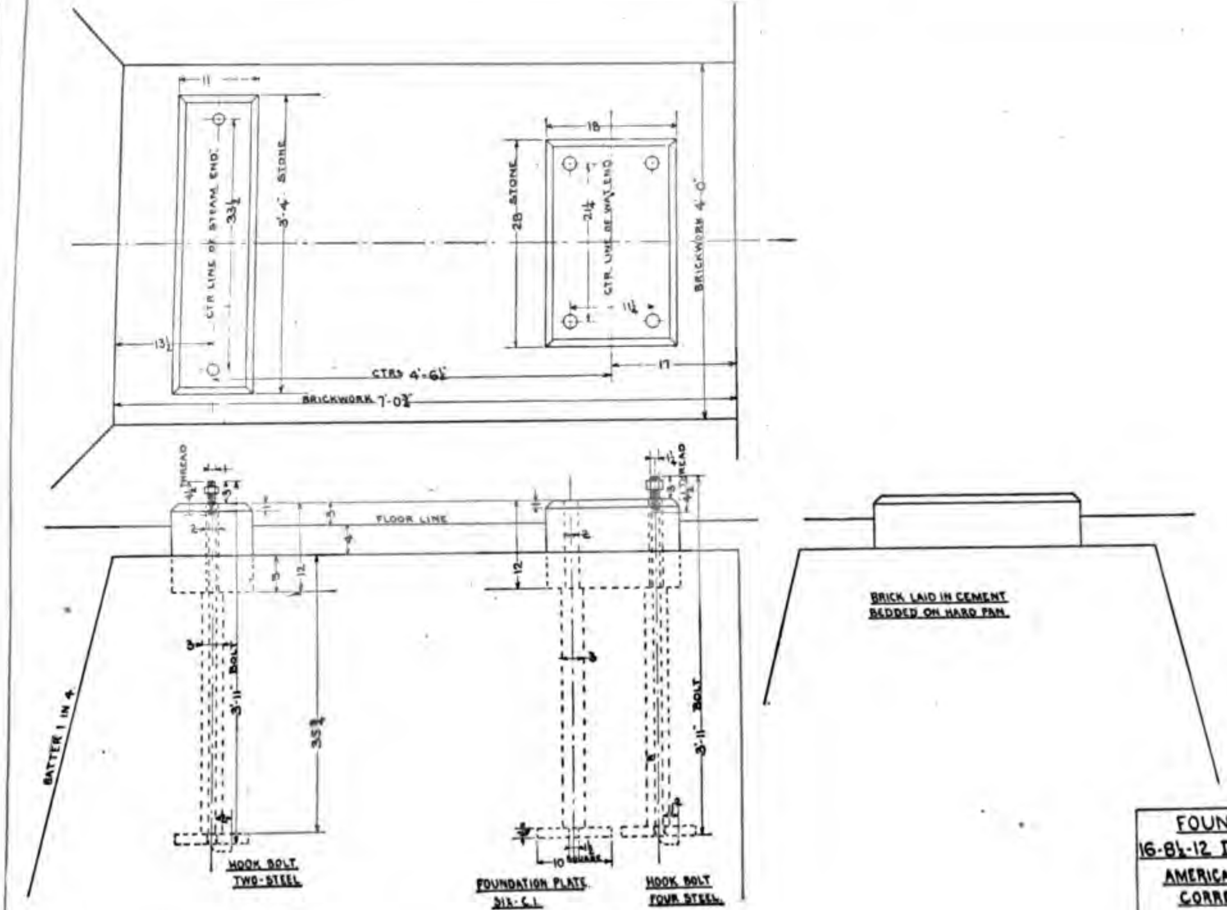
Plate M. Foundation. Pumps are often set directly upon a foundation of brick, but it makes a better job to bed stones, with surfaces dressed plane and true, into the main foundation, and rest the pump feet upon these stones. The simplest form of holding down bolts are shown on Plate M, a plain hook at the lower end, pulling up against a flat cast-iron plate, to distribute the pressure into the brickwork. These plates are of course bedded, and the bolts set as the foundation is built up. As the subsequent courses are laid some little space is left around the bolts, which may be afterwards filled with cement, thus making the bolts rigid with the foundation.

The water end of the foundation has no batter, because the suction pipe often drops vertically down from the end of the pump, and clearance is therefore necessary.

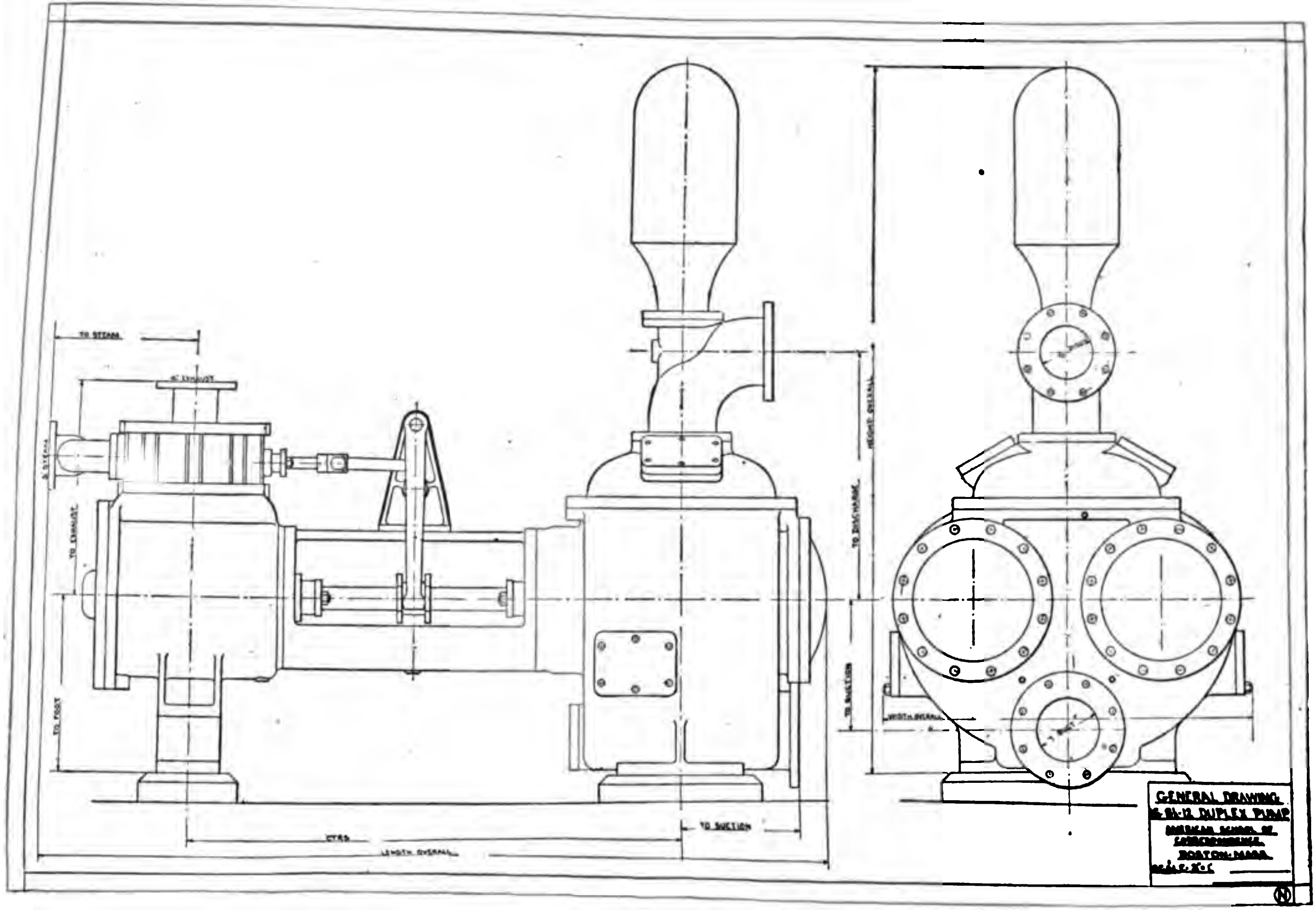
The floor line is placed 4 inches above the brickwork, to allow for the usual 1-inch top floor and 2-inch plank beneath, and still have a space left for shims to level the floor.

Plate N. General Drawing. This is an example of a plain, everyday shop drawing, to show the relation of parts and the extreme space occupied by the pump. A great deal of time can be needlessly wasted in producing a drawing of this character, by trying to make too faithful a picture. For example: If all the bolt heads were put in, it is safe to say that several hours' extra time would be required for this one item alone. But the drawing would be no better for shop use. Hence all bolt heads and nuts have been left out, except when necessary to show clearance.

Shade lines have been put on for no special reason except



FOUNDATION
16-8½-12 DUPLEX PUMP
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE - 1½" = 1'


TO STEAM
6" EXHAUST
3" STEAM
TO EXHAUST
TO FOOT
CTRS.
LENGTH OVERALL
TO SUCTION
TO DISCHARGE
HEIGHT OVERALL
TO SUCTION
WIDTH OVERALL
GENERAL DRAWING
6-8½-12 DUPLEX PUMP
AMERICAN SCHOOL OF
CORRESPONDENCE
BOSTON, MASS.
SCALE [illegible]
N

that if they are desirable on any drawing they are especially desirable on a general drawing, where one part overlaps another, as they make it easier to pick out and separate one surface from another. Some lines are shaded in this drawing which are not strictly sharp edges. It is held, however, that the rounding of a corner ought not to destroy its character as an edge casting a shadow, and such lines are treated accordingly.

An assembled or general drawing of this character should be laid out strictly from the dimensions shown by the details. It thus serves a valuable purpose in checking up figures, and showing whether or not the parts will go together. The method or character of the work in no respect differs from that suggested for the detail drawings.

If a scale of 3 inches = 1 foot be used, the size of sheet must be 24 inches × 36 inches. The student, however, will perhaps find it easier to use a scale of $1\frac{1}{2}$ inches = 1 foot, in which case the ordinary size, 18 inches × 24 inches, will suffice. For such a small scale it will be found undesirable to attempt to put in any very small fillets and corners, although those that can be readily handled by the ordinary bow pen ought not to be omitted. As a matter of fact, the expert draftsman either leaves the corners sharp, as suggested, or puts in the smallest curves freehand.

Order Sheets. Any set of drawings is incomplete unless in connection with it a statement is made in tabular form of the complete make-up of the machine. An infinite variety of ways exists for making the specifications. Sometimes the tabulated data are placed on the general drawing. Most often, however, printed blanks are provided, usually of bond paper, arranged with special reference to the individual shop system and methods of handling work; these blanks are filled in by the draftsman, indexed and filed as a part of the set of drawings. They can be blue printed for use in the shops the same as a drawing. From these sheets stock is ordered, checked off, and watched in its process of manufacture.

Order sheets are indispensable in any well-ordered shop. Hence they are illustrated on pages 59, 60, 61 and 62 as the final step in the set of pump drawings. They are made as simple as possible, and are not intended to fit any special shop system. As previously stated, the exact form and method of classification can be determined only when the shop conditions are known.

The student, having carefully followed through the preceding pages, must not think that he is master of pump construction, for even the type illustrated has been but touched upon. The object of the detailed discussion is to get the student in close touch with the spirit of construction, to make his drawings real serious work. It is hoped that the student will work just as though a machine were to be built from his drawings, and built to sell at a profit. Only in this way can advanced work in mechanical drawing be of benefit to him, for after becoming expert in the use of the instruments, no other advance is possible except advance in *thought.*

DATE.
MAY 20, 1902.

AMERICAN SCHOOL OF CORRESPONDENCE
AT
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

TYPE.
INSIDE PLUNGER.

LIST OF CASTINGS

FOR

16–8½–12 DUPLEX PUMP.

No. Wanted.	Name.	Drawing No.	Patt. or Piece No.	Material.	Remarks.
2	Steam Cylinder	G		C. I.	R. & L.
2	Steam Cylinder Head	G		C. I.	
2	Steam Chest	F		C. I.	
2	Steam Chest Cover	F		C. I.	
2	Slide Valve	F		C. I.	
1	Steam Pipe	F		C. I.	
1	Exhaust Tee	F		C. I.	
2	Valve Steam Gland	F		C. I.	
2	Piston	E		C. I.	
8	Piston Pipe Plug, 1½″	E		C. I.	
4	Piston Packing Ring	E		C. I.	
2	Spool	E		C. I.	
1	Steam Cylinder Cricket	L		C. I.	
2	Steam Cylinder Stuffing Box	L		C. I.	
2	Water Cylinder Stuffing Box	L		C. I.	
4	Piston Rod Gland	L		C. I.	
1	Valve Lever Bracket	L		C. I.	
2	Yoke	L		C. I.	R. & L.
1	Short Rocker Arm	K		C. I.	
1	Long Rocker Arm	K		C. I.	
2	Valve Stem Link	K		C. I.	
1	Water Cylinder	J		C. I.	
2	Water Cylinder Head	J		C. I.	
3	Hand Hole Cover	J		C. I.	
1	Water Cylinder Cap	I		C. I.	
1	Air Chamber	I		C. I.	
1	Discharge Ell	I		C. I.	
2	Plunger	H		C. I.	
2	Plunger Bushing	H		Brass	
2	Clamp Ring	H		C. I.	
32	Valve Stem	H		Brass	
32	Valve Cover	H		Brass	
32	Valve Seat	H		Brass	
6	Foundation Plate	M		C. I.	

DATE.
MAY 20, 1902.

AMERICAN SCHOOL OF CORRESPONDENCE
AT
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

TYPE.
INSIDE PLUNGER.

LIST OF STEEL AND MISCELLANEOUS PARTS
FOR
16–8½–12 DUPLEX PUMP.

No. Wanted.	Name.	Drawing No.	Patt. or Piece No.	Material.	Remarks.
2	Valve Steam Head	E		St.	Drop Forging.
2	Piston Rod	E		C. R. S.	
2	Valve Stem	E		St.	
1	Long P. R. Lever	K		St.	Forging
1	Short P. R. Lever	K		St.	Forging
1	Upper Rocker Shaft	K		St.	
1	Lower Rocker Shaft	K		St.	
2	Rocker Arm Pin	K		St.	
2	Link Pin	K		St.	
1	Long P. R. Lever Key	K		St.	Drop Forging
1	Short P. R. Lever Key	K		St.	Drop Forging
2	Rocker Arm Key	K		St.	Drop Forging
32	Valve Spring	H		Brass wire	Spring Temper
32	Valve	H		Rubber	Medium

DATE.
MAY 20, 1902.

AMERICAN SCHOOL OF CORRESPONDENCE
AT
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

TYPE.
INSIDE PLUNGER.

LIST OF BOLTS, NUTS AND PINS

FOR

16—8½—12 DUPLEX PUMP.

No. wanted.	Name.	Drawing No.	Patt. or Piece No.	Material.	Remarks.
24	Cylinder Head Stud ⅞ x 3¼	G		St.	
20	Steam Chest Stud ¾ x 8¾	G		St.	
4	Valve Stem Gland Stud ⅝ x 4¾	F		St.	
8	Piston Rod Gland Stud ¾ x 4	L		St.	
24	Water Cylinder Head Stud ⅞ x 3¼	J		St.	
12	Clamp Ring Stud ¾ x 4½	J		Tobin bz.	
24	Water Cylinder Cap Stud 1 x 3½	J		St.	
18	Hand Hole Cover Stud ⅝ x 2½	J		St.	
12	Hand Hole Cover Stud ⅝ x 2½	I		St.	
8	Exhaust Tee Tap Bolt ⅝ x 1¾	G		St.	
16	Yoke Tap Bolt ¾ x 2	G		St.	
8	Steam Cyl. Stf. Box Tap Bolt ¾ x 1¾	G		St.	
8	Steam Pipe Tap Bolt ⅝ x 1⅛	F		St.	
4	Valve Lever Bracket Tap Bolt ⅝ x 1¾	F		St.	
4	Steam Cyl. Cricket Tap Bolt 1 x 2¼	F		St.	
16	Yoke Tap Bolt ¾ x 2	J		St.	
8	Water Cyl. Stf. Box Tap Bolt ¾ x 1¾	J		St.	
8	Discharge Ell Tap Bolt ¾ x 2	I		St.	
4	Air Chamber Tap Bolt ¾ x 2	I		St.	
2	Hook Bolt (special) 1 x 3′ — 11″	M		St.	
4	Hook Bolt (special) 1¼ x 3′ — 11″	M		St.	
1	Eye Bolt Standard 1″	I		St.	
34	Standard Nut ⅝			St.	
44	Standard Nut ¾			St.	
36	Standard Nut ⅞			St.	
26	Standard Nut 1			St.	
4	Standard Nut 1¼			St.	
4	Standard Nut 2			St.	
8	Special Valve Stem Nut 1			St.	¾ Thick
4	Piston Rod Split Pin ¼ x 2	E		St.	
2	Spool Taper Pin No. 10 Morse Taper	E		St.	4″ long
4	Valve Bracket Dowel Pin ½ x 2	L		St.	

DATE.
MAY 20, 1902.

AMERICAN SCHOOL OF CORRESPONDENCE
AT
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

TYPE.
INSIDE PLUNGER.

LIST OF SPECIAL FITTINGS, WRENCHES, ETC.

FOR

16–8½–12 DUPLEX PUMP.

No. Wanted.	Name.		Drawing No.	Patt. or Piece No.	Material.	Remarks.
4	Drip Cock	$\frac{1}{8}$"	G			
1	Drip Cock	$\frac{1}{4}$"	F			
2	Drip Cock	$\frac{1}{4}$"	L			
4	Oil Cup	$\frac{1}{4}$"	L			
5	Drip Cock	$\frac{3}{8}$"	J			
1	Relief Valve	$1\frac{1}{4}$"	I			175 lbs. pressure
1	Standard Fork Wrench	$\frac{5}{8}$"				
1	Standard Fork Wrench	$\frac{3}{4}$"				
1	Standard Fork Wrench	$\frac{7}{8}$"				
1	Standard Fork Wrench	1"				
1	Socket Wrench	$\frac{3}{4}$"				12" handle
1	Valve Stem Fork Wrench					
1	Valve Stem Socket Wrench					

PLATES.

The plates of Part VI (A to N, inclusive,) are so arranged and described that the complete set may be made by the student. In case of insufficient time, the following short examination is prescribed. The student in any case should read carefully all the text, and follow the discussion of all the plates, whether or not actually produced by him.

PLATE XXV.

Reproduce Plate E (page 32), putting on shade lines. Size of plate to be 18 inches × 24 inches, trimmed, with border $\frac{1}{2}$ inch from edge; size within border line, 17 inches × 23 inches.

PLATE XXVI.

Make steam end layout, Plate A (page 10). This is to be done accurately *in pencil.* Size same as Plate XXV

PLATE XXVII.

Make drawing of steam cylinder, Plate G (page 18), complete, with finish marks and dimensions. Do not put on shade lines.

PLATE XXVIII.

Draw valve motion layout, Plate C (page 36). This plate also is to be done accurately *in pencil.*

PLATE XXIX.

Make rough freehand sketches in pencil of parts on Plate L (page 42), putting on finish marks and dimensions. Use a medium pencil, and make sheet of regular size (18 inches × 24 inches). Arrangement to be similar to Plate L. Do not try to make the sketches to scale, but make them sufficiently complete so that detail drawings can be made from them.

PLATE XXX.

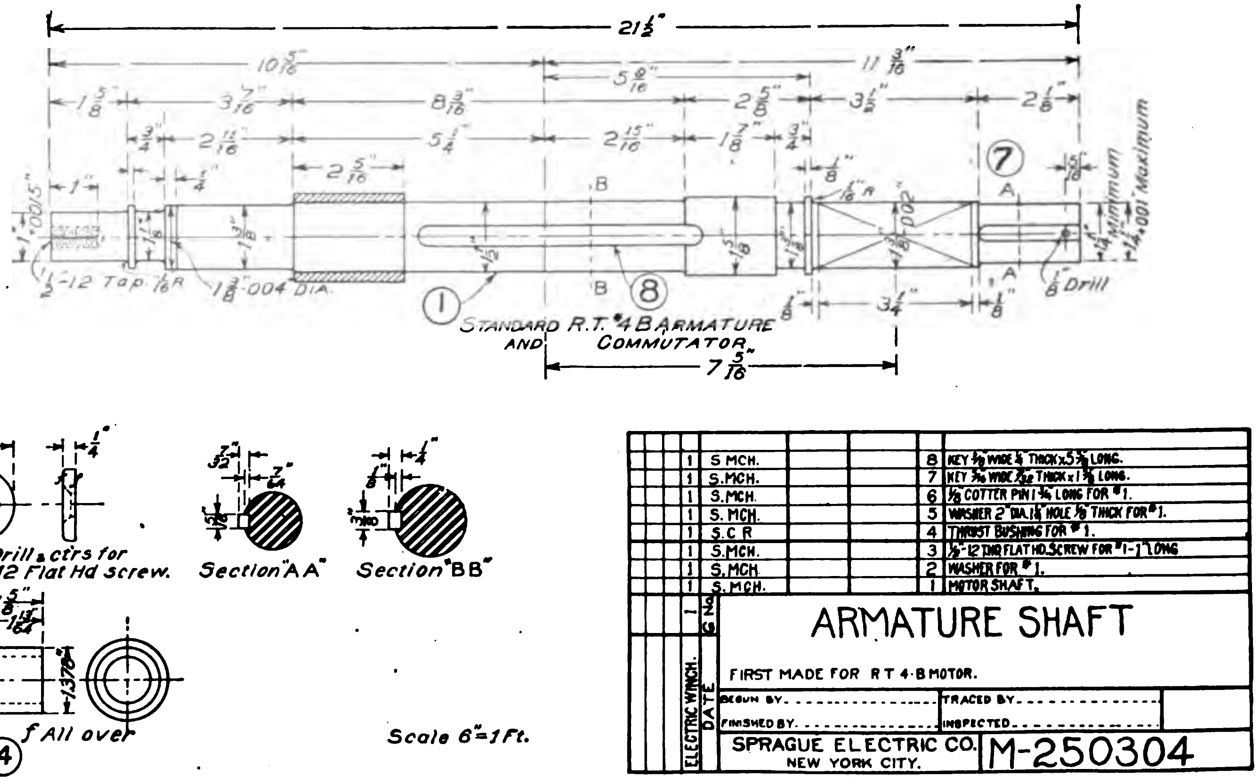
Make a tracing on tracing cloth of Plate E (page 32).

PLATE XXXI.

Make general drawing Plate N. Use scale of $1\frac{1}{2}'' = 1'$, or $3'' = 1'$.

PLATE XXXII (Optional).

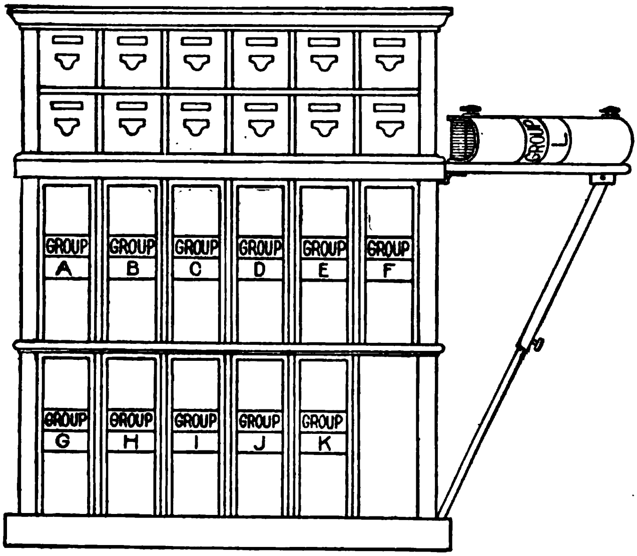
Make drawing of water cylinder, Plate J (page 48).



TYPICAL WORKING DRAWING OF ARMATURE-SHAFT MADE IN THE DRAFTING ROOM OF THE SPRAGUE ELECTRIC CO.

DRAFTING ROOM ORGANIZATION

It is always of interest to manufacturing engineers when anything additional to the subject of systematic commercial drawing is brought to their attention. The development of a universal drawing process or system sufficiently general to meet all practical needs of manufacturing concerns, will likely come about at some future time, and this may form part of the engineering curriculum in schools and colleges, much the same as bookkeeping has been taught in commercial colleges for years.



Cabinet Set Forming Portion of Part Drawing System.
Comprises, in the upper part, a card index; and in the lower part, a bound blue-print file containing records and drawings of finished material or parts.

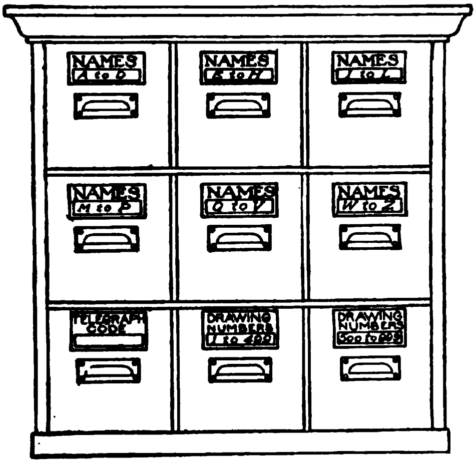
The present is a period of standard manufacturing, and the tendency toward standardization is naturally growing, thereby creating a demand for different methods from those employed where the articles of manufacture were unsettled and undergoing ceaseless and uncertain changes. It is of course true that improvements in standard lines of manufacture will be continually required, but these changes take the form of revisions, to a greater or less extent, of designs that previously have been reduced to a fairly definite basis. The drawing office is where things are built on paper, and it is subject to a change of methods when approaching standard manufacturing as much as are the factory departments doing actual construction work.

To illustrate—the following list brings out a few of the essential features that usually appear for consideration in conducting a drawing office for developing specialized products:

1. The desirability of one uniform size of drawing sheet for all drawings.
2. The necessity of drawing parts independently of any mechanism to which they may belong, for the reason that any part is likely to be found available for use in the manufacture of a number of different mechanisms.
3. The necessity of knowing and being able to identify all parts by symbols.
4. A suitable system of drawing parts.
5. The necessity of classifying parts into groups or collections and grading them according to the primary operations that are required in forming them, or according to their elementary nomenclature or design.
6. As parts and constructional views cannot be shown on the same drawing, some means must be employed by draftsmen to specify and identify the parts that are involved in a general drawing.
7. Once typical general drawings have been made, how is it possible to avoid the delay and expense that are inevitably attached to redrawing practically the same thing for the many modifications arising from time to time?

Standard manufacturing may be looked upon as differing from special or miscellaneous manufacturing inasmuch as, under the former, parts take a systematic course from one set of machines to another until they finally become finished material.

The work of constructing or assembling the various parts to form a finished product, involves an entirely separate office. General drawings are of service principally where assembling is done, while the part-producing departments, on the other hand, require but those drawings that give working dimensions of parts. It is therefore seen that, so far as the uses of drawings are concerned, there is no occasion for showing parts associated with the finished mechanism, even were it possible to do so. Moreover, the fact that any single part may be found available in a variety of places, suggests the principle of showing parts collectively as a body of things having individual entity; and, if they are to be thought of as associated in some manner, it seems preferable to regard then as some group or collection. Where numerous parts are involved, the idea of drawing each of them separately on an independent sheet—which is a very general practice—incurs the disadvantage of a great collection of small sheets, whereas, by the use of a no less convenient but larger sheet, their number may be materially reduced by placing five to seven or more part views on the one drawing, arranging them, of course, according to their respective groups. One uniform size of drawing sheet can be used, which is of great advantage in filing and binding blue-prints for circulation.



Section of Cabinet Set for Finished Products.

Contains material record list cards arranged alphabetically under names of mechanism, general view drawing numbers, and telegraph code for drawing office reference.

Symbols. Symbols for parts usually take the form of numbers, with characters or letters annexed which signify the association of the parts. Each and every part is numbered in numerical sequence; and in the present instance, letters may be annexed which signify the group of which the part is a member, and the groups established according to the basic operations by which they are first formed. This is illustrated by the following list:

Cast Metal Parts.—*Group Symbol,* A.

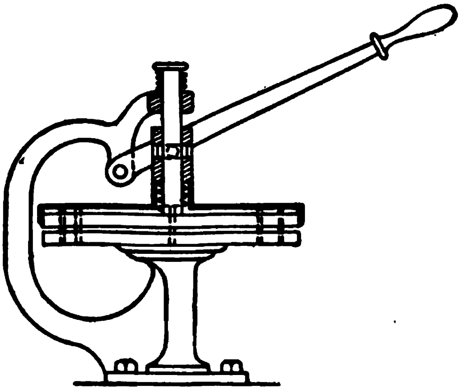
All parts originally formed by moulding or casting, and which are not magnets.

Sheet Metal Parts, Die Formed, etc.—*Group Symbol, B.*

All parts originally formed from sheet metal by puncturing, shearing, spinning, stamping, etc., and which are not magnets.

Machine-Formed Parts.—*Group Symbol, C.*

All metal parts machine-formed in lathes, shapers, milling machines, screw machines, threading and tapping machines, etc., such parts being made from raw material in the shape of rods, bars, tubes, plates, etc., and which are not magnets.



Trimmer and Punch-Press.

One advantage in employing a single standard size of sheet for part drawings is that the blue-prints can be machine-trimmed by a lightly constructed punch-press, which at the same time will pierce the holes for binding.

Magnets.—*Group Symbol, D.*

Cores of electro-magnets, armatures, pole-pieces, and shoes, frames, permanent magnets, shells, yokes, etc.

Insulators.—*Group Symbol, E.*

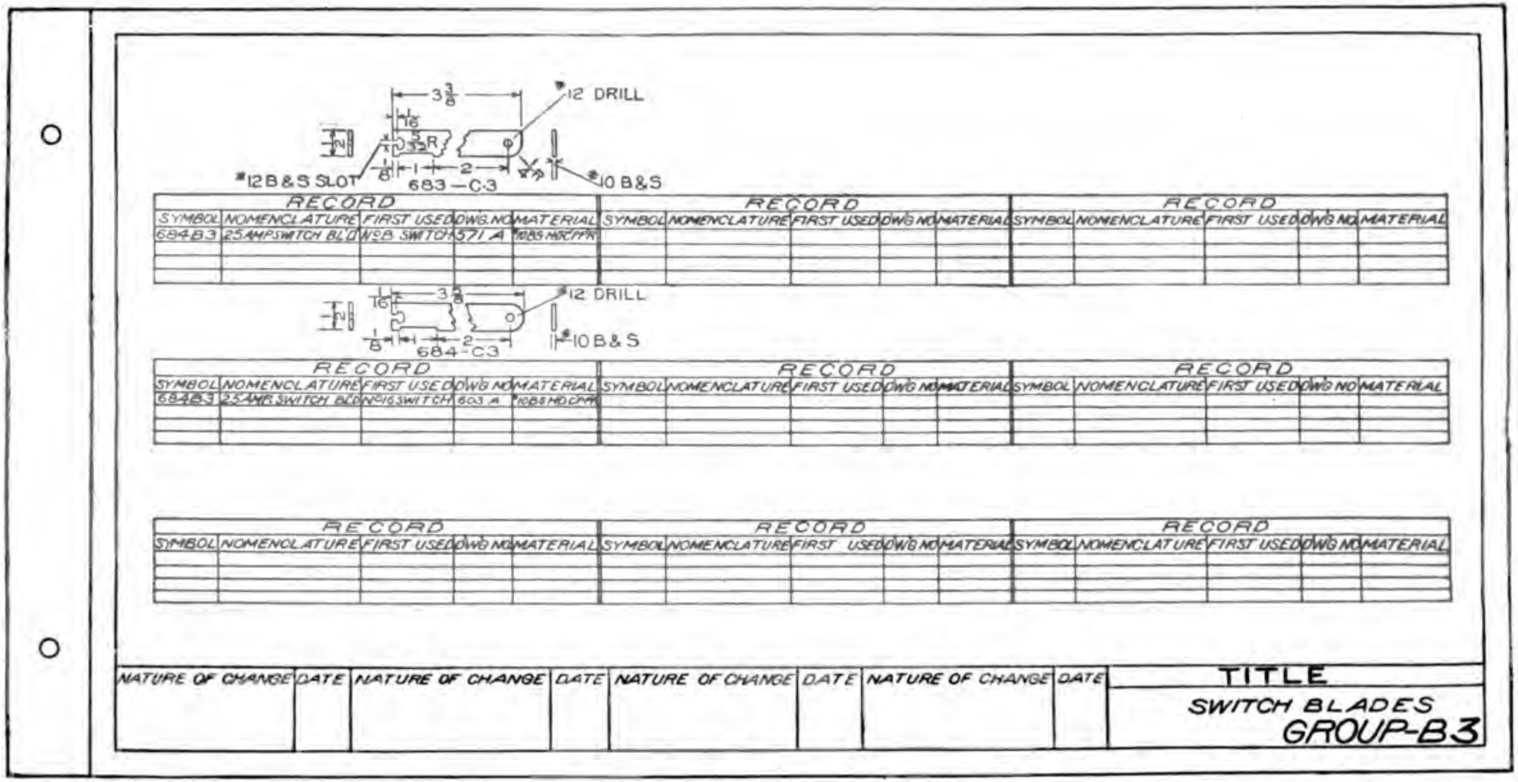
Tubes, strips, bushings, collars, handles, knobs, etc., of porcelain, glass, rubber, fiber, wood, paper; fabrics or parts made of other material for insulating purposes.

Coils.—*Group Symbol, F.*

Electro-magnet coils, resistance coils, condenser coils, or current-carrying coils of any nature; to comprise only the coil governed by its winding measurements, size of wire, number of turns, resistance, grade of insulation, and material wire is made of.

Forged or Wrought Parts.—*Group Symbol, G.*

All parts such as hooks, rivets, rings, wire springs, which are made of wire



RECORD					RECORD					RECORD				
SYMBOL	NOMENCLATURE	FIRST USED	DWG. NO	MATERIAL	SYMBOL	NOMENCLATURE	FIRST USED	DWG NO	MATERIAL	SYMBOL	NOMENCLATURE	FIRST USED	DWG NO	MATERIAL
684 B3	25 AMP SWITCH BL'D	No. 8 SWITCH	571 A	#10 BS HD CPR										

RECORD					RECORD					RECORD				
SYMBOL	NOMENCLATURE	FIRST USED	DWG NO	MATERIAL	SYMBOL	NOMENCLATURE	FIRST USED	DWG NO	MATERIAL	SYMBOL	NOMENCLATURE	FIRST USED	DWG NO	MATERIAL
684 B3	25 AMP SWITCH BL'D	No. 16 SWITCH	603 A	#10 BS HD CPR										

RECORD					RECORD					RECORD				
SYMBOL	NOMENCLATURE	FIRST USED	DWG NO	MATERIAL	SYMBOL	NOMENCLATURE	FIRST USED	DWG NO	MATERIAL	SYMBOL	NOMENCLATURE	FIRST USED	DWG NO	MATERIAL

NATURE OF CHANGE	DATE	NATURE OF CHANGE	DATE	NATURE OF CHANGE	DATE	NATURE OF CHANGE	DATE	TITLE
								SWITCH BLADES GROUP-B3

Specimen Part Drawing.

and which are not resistance coils; supports, brackets, fixtures, and all such parts, which are hand-formed, forged, or wrought, and which are not magnets.

Wooden Parts.—*Group Symbol, H.*

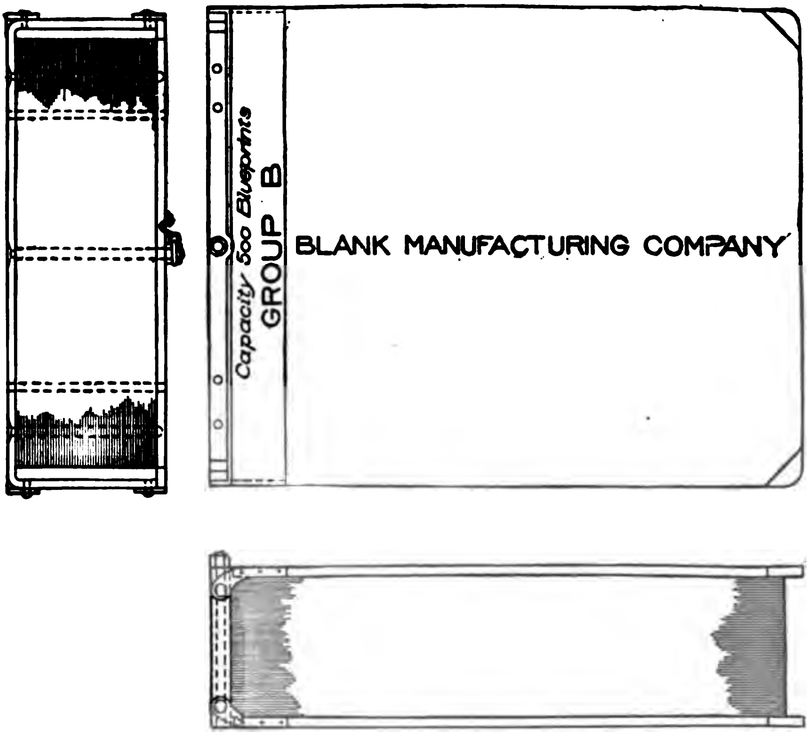
All wooden parts such as strips, pins, blocks, bases, moulding, ornaments, knobs, and brackets, which are not used for insulating purposes.

Compound Parts.—*Group Symbol, J.*

All parts consisting of more than one member, where it is necessary to join them to produce the part as completed material ready for assembling.

Purchased Articles.—*Group Symbol, K.*

All articles such as locks, door or drawer knobs, fixtures, receptacles, and ornaments, which are compound parts, being purchased, not manufactured.

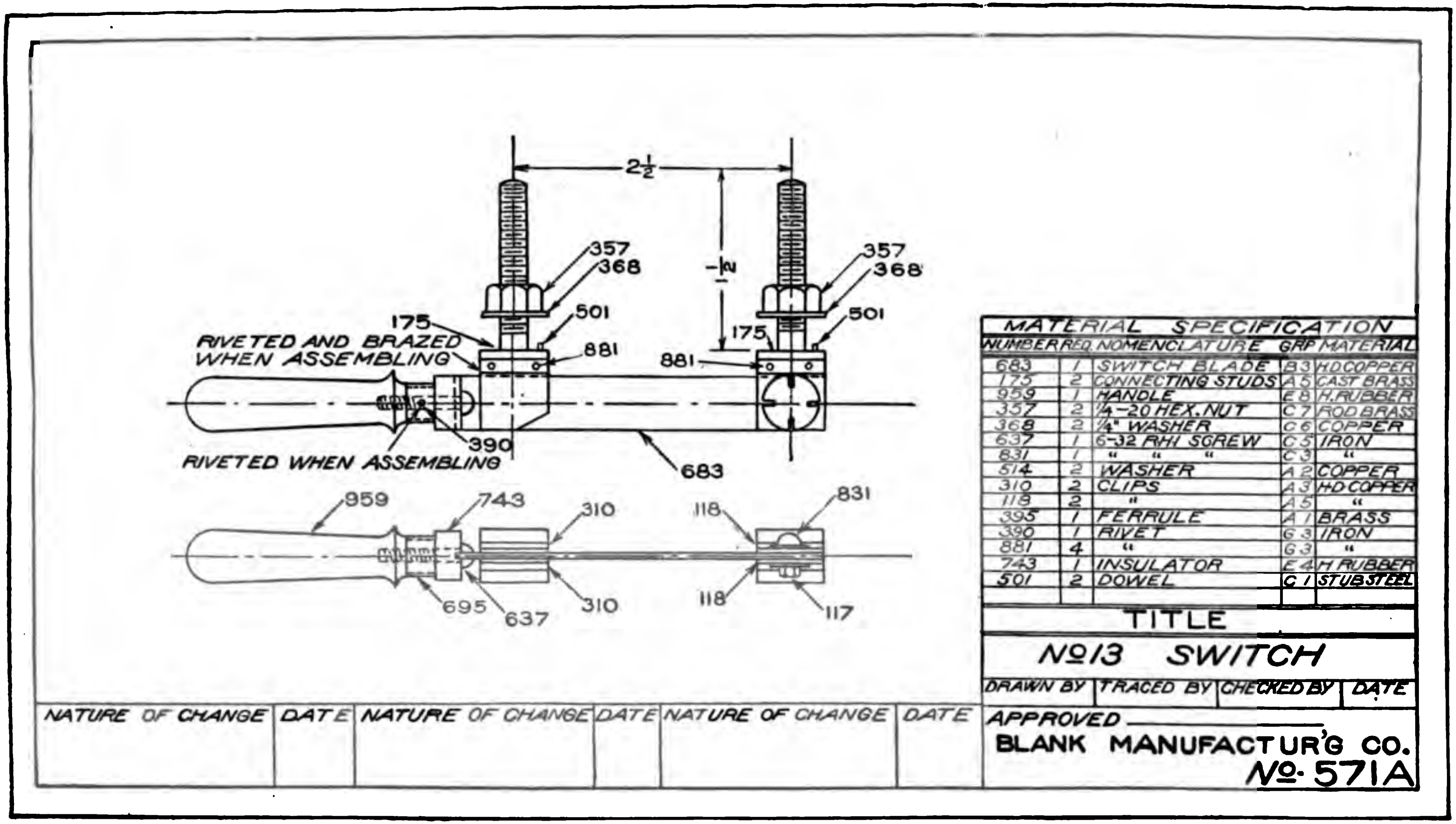


Detail of Blue-Print Binder Designed for Part Drawing System.

The binding clasp is independent of the covers, which are hinged and can be opened clear back, allowing free access to the prints. Any print can be removed or inserted at any point, and the binder conveniently rejoined by use of a combination key and guide.

The above classification enables an intelligent analysis of the parts to be made, placing them in their respective group order according to the directions or codifications given.

The drawings for each set exhibit a convenient number of part views, and the title of each sheet indicates the name and group



MATERIAL SPECIFICATION				
NUMBER	REQ.	NOMENCLATURE	GRP	MATERIAL
683	1	SWITCH BLADE	B 3	H.D.COPPER
175	2	CONNECTING STUDS	A 5	CAST BRASS
959	1	HANDLE	E 8	H.RUBBER
357	2	¼–20 HEX.NUT	C 7	ROD BRASS
368	2	¼" WASHER	C 6	COPPER
637	1	6–32 RHI SCREW	C 5	IRON
831	1	" " "	C 3	"
514	2	WASHER	A 2	COPPER
310	2	CLIPS	A 3	H.D.COPPER
118	2	"	A 5	"
395	1	FERRULE	A 1	BRASS
390	1	RIVET	G 3	IRON
881	4	"	G 3	"
743	1	INSULATOR	E 4	H.RUBBER
501	2	DOWEL	C 1	STUBSTEEL

TITLE

Nº13 SWITCH

DRAWN BY	TRACED BY	CHECKED BY	DATE

NATURE OF CHANGE	DATE	NATURE OF CHANGE	DATE	NATURE OF CHANGE	DATE

APPROVED ____________

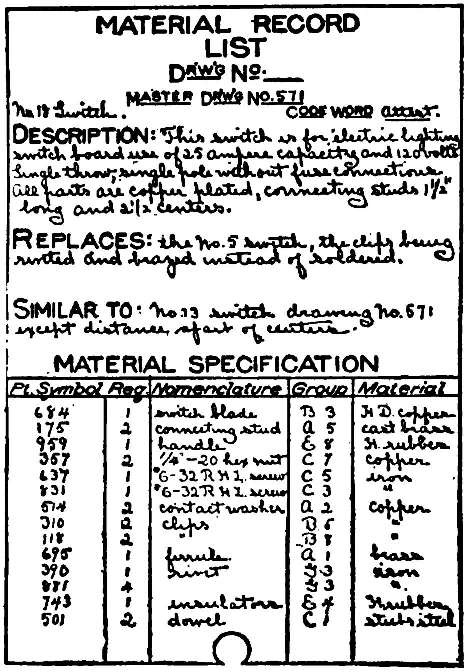
BLANK MANUFACTUR'G CO.

Nº. 571A

Specimen General View Drawing.

of the parts it contains. The sheets are numbered serially, beginning each set with *No. 1*, the number increasing by one for each successive addition. If, for illustration, the number portion of a part symbol is *368*, then, in order to signify its groups and still more its drawing, we might write *368A3*, which means that the part belongs to the *A* group (Cast Metal Parts), and is shown on the third drawing of the series.

Material Lists. Where articles of manufacture are repeatedly reproduced, as is the case in standard manufacturing, and where, as often occurs, they are, for various reasons, not made in all cases just exactly alike, there is involved the question as to how far the reproduction of general view drawings should be carried. A record at least is necessary of each article, just as it is built. The old and familiar but serviceable plan of employing material lists, developed through the use of part symbols, etc., serves admirably in combination with a typical or master drawing. Such lists enumerate the various parts used in each mechanism;



MATERIAL RECORD LIST

DRWG NO.___

No. 18 Switch. MASTER DRWG NO. 571 CODE WORD attest.

DESCRIPTION: This switch is for electric lighting switch board use of 25 ampere capacity and 120 volts. Single throw, single pole without fuse connections. All parts are copper plated, connecting studs 1½" long and 2½ centers.

REPLACES: the No. 5 switch, the clips being rivited and brazed instead of soldered.

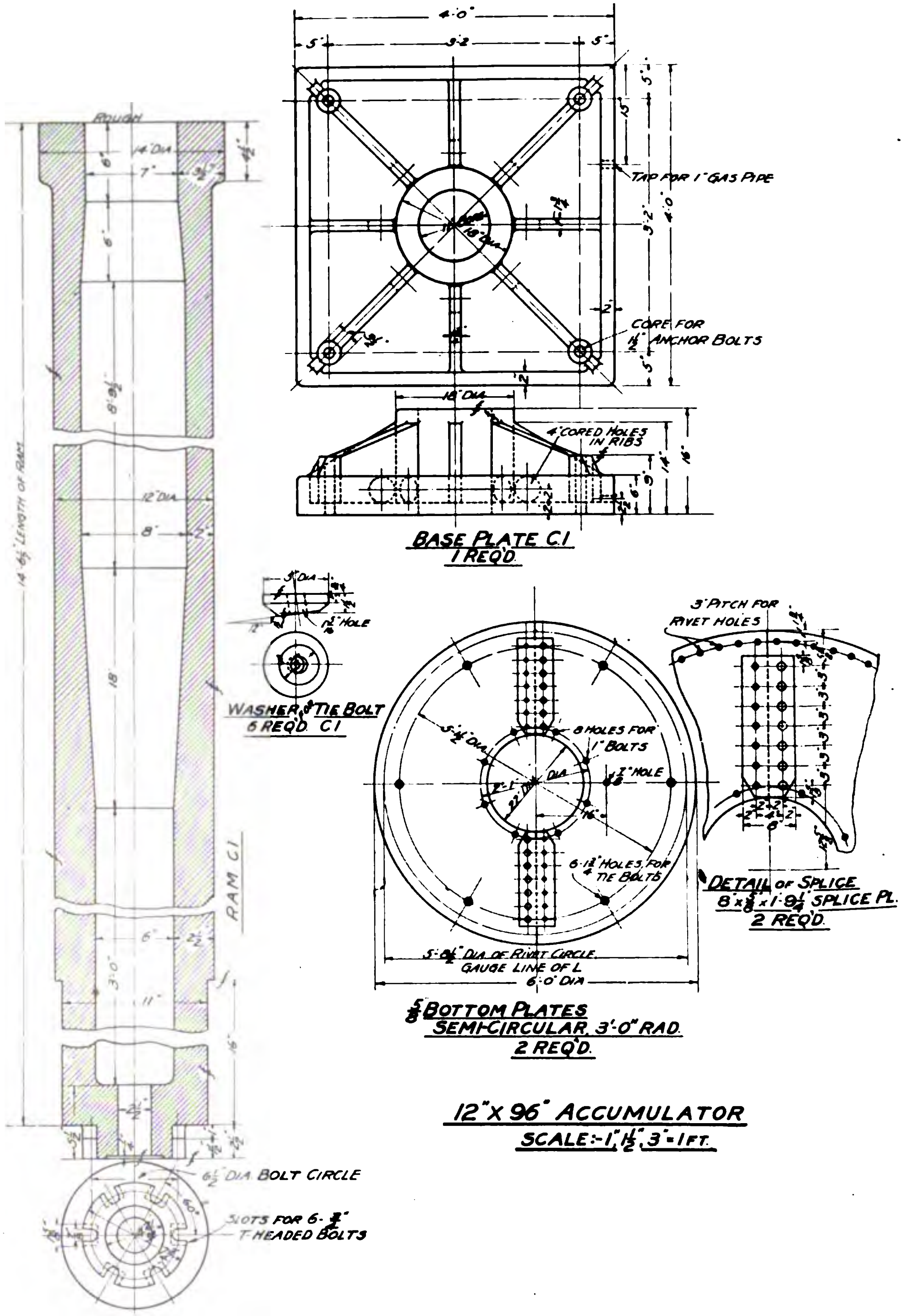
SIMILAR TO: No. 13 switch drawing No. 671 except distance apart of centers.

MATERIAL SPECIFICATION

Pt. Symbol	Req.	Nomenclature	Group	Material
684	1	switch blade	B 3	H. D. copper
175	2	connecting stud	A 5	cast brass
959	1	handle	E 8	H. rubber
357	2	¼"-20 hex nut	C 7	copper
637	1	°6-32 R. H. I. screw	C 5	iron
831	1	°6-32 R. H. I. screw	C 3	"
514	2	contact washer	A 2	copper
310	2	clips	B 6	"
118	2	"	B 8	"
695	1	ferrule	A 1	brass
390	1	rivet	E 3	iron
881	4		E 3	"
743	1	insulators	E 4	H. rubber
501	2	dowel	C 1	stub steel

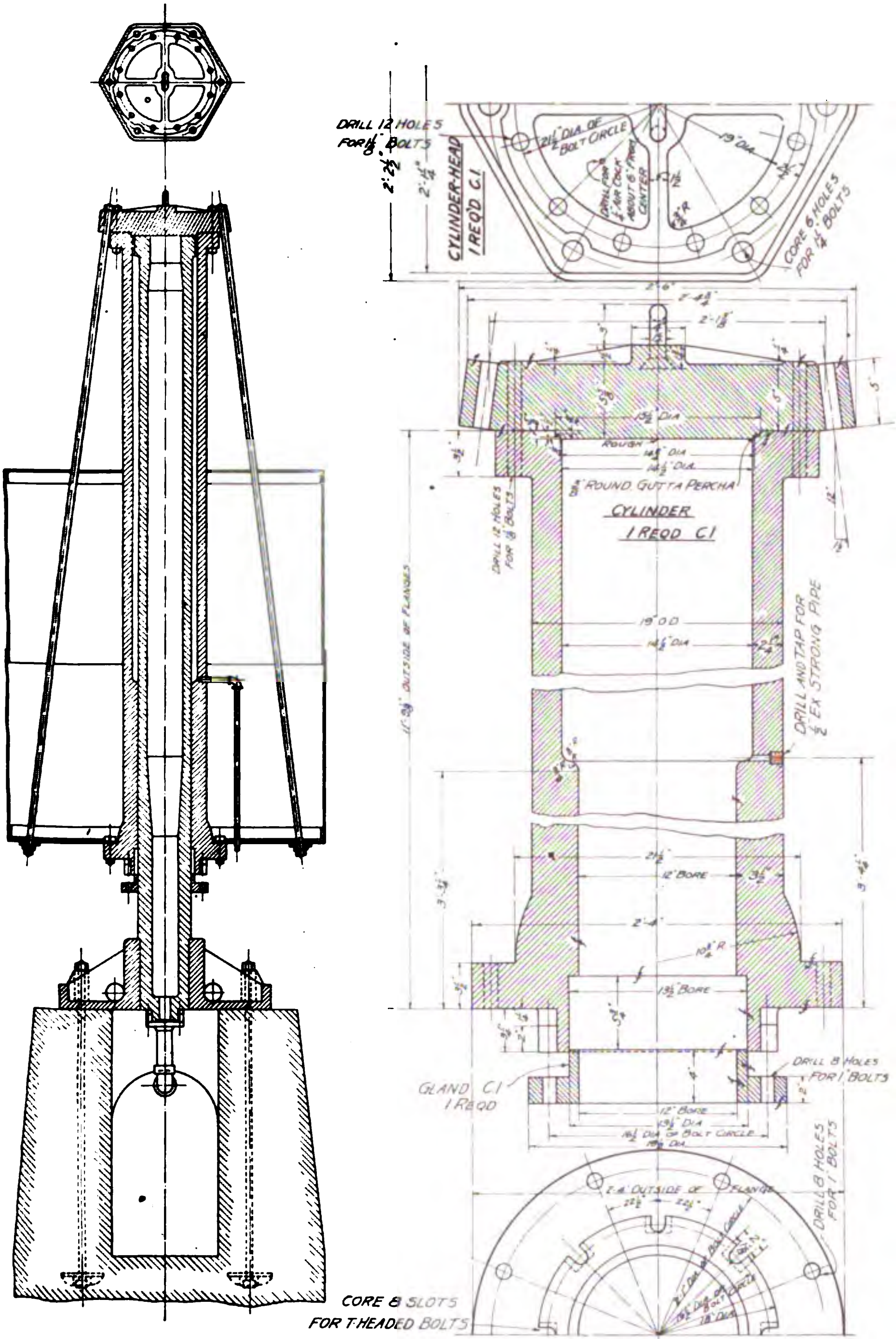
and when the construction changes, it must occur through a change in the combination of parts. Therefore, when the parts are enumerated there can be no mistake about the construction. To illustrate the use of such lists, let us assume the instance of a switch which is represented by a general drawing and called the "No. 13 Switch." Occasion later arises for a modification, say in the distance between stud centers. This will require a different blade, which is practically the only difference between the switches. The material record list shown in the accompanying illustration fully serves the purpose, as it reveals and identifies the component parts which, accompanied with a master drawing, furnish complete manufacturing data.

The "Material Specification" portion of the record card is in form a duplicate of the material list appearing in general drawings, being used for the same purpose—that of identifying the parts involved in a convenient tabulation.



HYDRAULIC ACCUMULATOR

Detail Drawing, to be used with Figure on Opposite Page

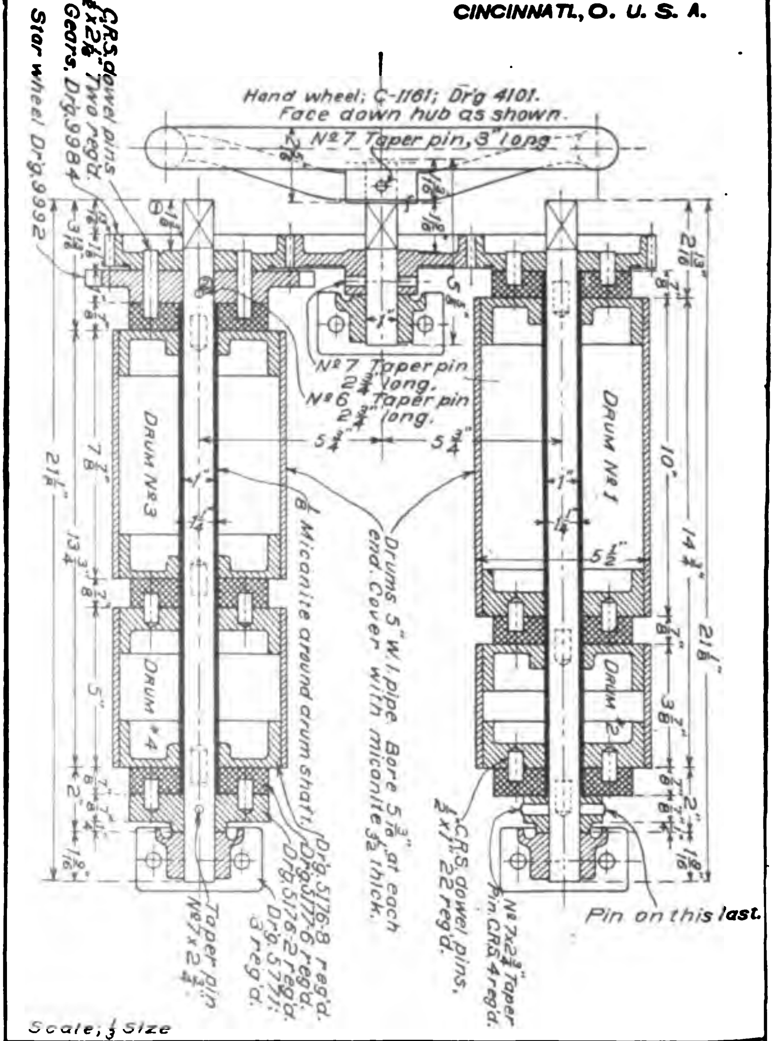


HYDRAULIC ACCUMULATOR

Assembled Drawing with Details on the Same Sheet

BULLOCK ELECTRIC MANUFACTURING CO.

CINCINNATI, O. U. S. A.


C.R.S. dowel pins $\frac{3}{8}$"x2$\frac{1}{16}$" Two reqd.
Gears. Drg. 9984
Star wheel Drg. 9992
Hand wheel; C-1161; Dr'g 4101.
Face down hub as shown.
Nº 7 Taper pin, 3" long
Nº 7 Taper pin 2$\frac{3}{4}$" long.
Nº 6 Taper pin 2$\frac{3}{4}$" long.
5$\frac{3}{4}$"
DRUM Nº 3
DRUM Nº 4
DRUM Nº 1
DRUM Nº 2
$\frac{1}{8}$ Micanite around drum shaft.
Drums 5" W.I. pipe. Bore 5$\frac{3}{16}$" at each end. Cover with micanite $\frac{1}{32}$ thick.
Drg. 5176-8 reqd.
Drg. 5177-6 reqd.
Drg. 5176-2 reqd.
Drg. 5771; 3 reqd.
Taper pin Nº 7 x 2$\frac{3}{4}$"
C.R.S. dowel pins $\frac{1}{4}$x1$\frac{3}{8}$" 22 reqd.
Nº 7 x 2$\frac{3}{4}$" Taper pin. C.R.S. 4 reqd.
Pin on this last.
21$\frac{1}{8}$"
13$\frac{3}{4}$"
7$\frac{7}{8}$"
10"
14$\frac{3}{4}$"
3$\frac{7}{8}$"
5"
1"
Scale; $\frac{1}{3}$ Size

9983-1 1-3-1902 ① Changed from 1$\frac{5}{8}$"; length over all, also changed to suit 9983	9983-2 4-1-1902 ② Changed from Nº 7 to Nº 6 taper pin A.B.W.	9983-3 4-1-1902 SHAFT AND ASSEMBLY OF DRUMS TYPE F. CONTROLLER.	Material.........No. Reqd. Tool No.................. Pat. No.................. .... Est. Weight.................. .... Draftsman A.B.W.

EXAMPLE OF SHOP DRAWING REPRODUCED FROM A BLUE PRINT BY THE COURTESY OF THE BULLOCK ELECTRIC MFG. CO.

MACHINE DESIGN.

PART I.

Definition. Machine Design is the art of mechanical thought development, and specification.

It is an art, in that its routine processes can be analyzed and systematically applied. Proficiency in the art positively cannot be attained by any "short cut" method. There is nothing of a spectacular nature in the methods of Machine Design. Large results cannot be accomplished at a single bound, and success is possible only by a patient, step-by-step advance in accordance with well-established principles.

"Mechanical thought" means the thinking of things strictly from their mechanical side; a study of their mechanical theory, structure, production, and use; a consideration of their mechanical fitness as parts of a machine.

"Mechanical development" signifies the taking of an idea in the rough, in the crude form, for example, in which it comes from the inventor, working it out in detail, and refining and fixing it in shape by the designing process. Ideas in this way may become commercially practicable designs.

"Mechanical specification" implies the detailed description of designs, in such exact form that the shop workmen are enabled to construct completely and put in operation the machines represented in the designs.

The object of Machine Design is the creation of machinery for specific purposes. Every department of a manufacturing plant is a controlling factor in the design and production of the machines built there. A successful design cannot be out of harmony with the organized methods of production. Hence in the high development of the art of Machine Design is involved a knowledge of the operations in all the departments of a manufacturing plant. The student is therefore urged not only to familiarize himself with the direct production of machinery, but to study the relation thereto of the allied commercial departments

He should get into the spirit of business at the start, get into the shop atmosphere, execute his work just as though the resulting design were to be built and sold in competition. He should visit shops, work in them if possible, and observe details of design and methods of finishing machine parts. In this way he will begin to store up bits of information, practical and commercial, which will have valuable bearing on his engineering study.

The labor involved in the design of a complicated automatic machine is evidenced by the designer's wonderful familiarity with its every detail as he stands before the completed machine in operation and explains its movements to an observer. The intricate mass of levers, shafts, pulleys, gears, cams, clutches, etc., etc., packed into a small space, and confusing even to a mechanical mind, seems like a printed book to the designer of them.

This is so because it is a familiar journey for the designer's mind to run over a path which it has already traversed so many times that he can see every inch of it with his eyes shut. Every detail of that machine has been picked from a score or more of possible ideas. One by one, ideas have been worked out, laid aside, and others taken up. Little by little, the special fitness of certain devices has become established, but only by patient, careful consideration of others, which at first seemed equally good.

Every line, and corner, and surface of each piece, however small that piece may be, has been through the refining process of theoretical, practical, and commercial design. Every piece has been followed in the mind's eye of its designer from the crude material of which it is made, through the various processes of finishing, to its final location in the completed machine; thus its bodily existence there is but the realization of an old and familiar picture.

What wonder that the machine seems simple to the designer of it! As he looks back to the multitude of ideas invented, worked out, considered and discarded, the machine in its final form is but a trifle. It merely represents a survival of the fittest.

No successful machine, however simple, was ever designed that did not go through this slow process of evolution. No machine ever just simply happened by accident to do the work for which it is valued. No other principle upon which the successful design of machinery depends is so important as this careful, patient consideration of detail. A machine is seldom unsuccessful because some main point of construction is wrong. The principal features of a machine are usually the easiest to determine. It is a failure because some little detail was overlooked, or hastily considered, or allowed to be neglected, because of the irksome labor necessary to work it out properly.

There is no task so tedious, for example, as the devising of the method of lubricating the parts of a complicated machine. Yet there is no point of design so vital to its life and operation as an absolute assurance of an adequate supply of oil for the moving parts at all times and under all circumstances. Suitable means often cannot be found, after the parts are together, hence the machine goes into service on a risky basis, with the result, perhaps, of early failure, due to "running dry." Good designers will not permit a design to leave their hands which does not provide practically automatic oiling, or at least such means of lubrication that the operator can offer no excuse for neglecting to oil his machine. This is but a single illustration of many which might be presented to impress the definite and detail character necessary in work in Machine Design.

Relation. The relation which Machine Design should correctly bear to the problems that it seeks to solve, is twofold; and there are, likewise, two points of view corresponding to this twofold relation, from which a study of the subject should be traced. Neither of these can be discarded and an efficient mastery of the art attained. These points are—

I. Theory.
II. Production.

I. *Theory.* From this point of view, Machine Design is merely a skeleton or framework process, resulting in a representation of ideas of pure motion, fundamental shape, and ideal proportion. It implies a working knowledge of physical and mathematical laws. It is a strictly scientific solution of the problem at hand, and may be based purely on theory which has been reasoned out by calculation or deduced from experiment. This is the only sure foundation for intelligent design of any sort.

But it is not enough to view the subject from the standpoint of theory alone. If we stopped here we should have nothing but mechanisms, mere laboratory machines, simply structures of ingenuity and examples of fine mechanical skill. A machine may be correct in the theory of its motions; it may be correct in the theoretical proportions of its parts; it may even be correct in its operation for the time being; and yet its complication, its misdirected and wasteful effort, its lack of adjustment, its expensive and irregular construction, its lack of compactness, its difficulty of ready repair, its inability to hold its own in competition—any of these may throw the balance to the side of failure. Such a machine, commercially considered, is of little value. No shop will build it, no machinery house will sell it, nobody will buy it if it is put on the market.

Thus we see that, aside from the theoretical correctness of principle, the design of a machine must satisfy certain other exacting requirements of a distinctly business nature.

II. *Production.* From this point of view, Machine Design is the practical, marketable development of mechanical ideas. Viewed thus, the theoretical, skeleton design must be so clothed and shaped that its production may be cheap, involving simple and efficient processes of manufacture. It must be judged by the latest shop methods for exact and maximum output. It must possess all the good points of its competitor, and, withal, some novel and valuable ones of its own. In these days of keen competition it is only by carefully studied, well-directed effort toward rapid, efficient, and, therefore, cheap production that any machine can be brought to a commercial basis, no matter what its other merits may be. All this must be thought of and planned for in the design, and the final shapes arrived at are quite as much a result of this second point of view as of the first.

As a good illustration of this, may be cited the effect of the present somewhat remarkable development of the so-called "high speed" steels. The speeds and feeds possible with tools made of these steels are such that the driving power, gearing, and feed mechanism of the ordinary lathe are wholly inadequate to the demands made upon them when working the tool to its limit. This means that the basis of design as used for the ordinary tool steel will not do, if the machine is expected to stand up to the cuts possible with the new steels. Hence, while the old designs were right for the old standard, a new one has been set, and a thorough revision on a high-speed basis is imminent, else the market for them as machines of maximum output will be lost.

From these definitions it is evident that the designer must not only use all the theory at his command, but must continually inform himself on all processes and conditions of manufacture, and keep an eye on the tendency of the sales markets, both of raw material and the finished machinery product. This is what in the broadest sense is meant by the term "Mechanical Thought," thought which is directed and controlled, not only by theoretical principle but by closely observed practice. From the feeblest pretenders of design to those engineers who consummate the boldest feats and control the largest enterprises, the process which produces results is always the same. Although experience is necessary for the best mechanical judgment, yet the student must at least begin to cultivate good mechanical sense very early in his study of design.

Invention. Invention is closely related to Machine Design, but is not design itself. Whatever is invented has yet to be designed. An invention is of little value until it has been refined by the process of design.

Original design is of an inventive nature, but is not strictly invention. Invention is usually considered as the result of genius, and is announced in a flash of brilliancy. We see only the flash, but behind the flash is a long course of the most concentrated brain effort. Inventions are not spontaneous, are not thrown off like sparks from the blacksmith's anvil, but are the result of hard and applied thinking. This is worth noting carefully, for the same effort which produces original design may develop a valuable invention. But there is little possibility of inventing anything except through exhaustive analysis and a clear interpretation of such analysis.

Handbooks and Empirical Data. The subject matter in these is often contradictory in its nature, but valuable nevertheless. Empirical data are data for certain fixed conditions and are not general. Hence, when handbook data are applied to some specific case of design, while the information should be used in the freest manner, yet it must not be forgotten that the case at hand is probably different, in some degree, from that upon which the data were based, and unlike any other case which ever existed or will ever again exist. Therefore the data should be applied with the greatest discretion, and when so applied will contribute to the success of the design at least as a check, if not as a positive factor.

The student should at the outset purchase one good handbook, and acquire the habit of consulting it on all occasions, checking and comparing his own calculations and designs therefrom. Care must be taken not to become tied to a handbook to such an extent that one's own results are wholly subordinated to it. Independence in design must be cultivated, and the student should not sacrifice his calculated results until they can be shown to be false or based on false assumption. Originality and confidence in design will be the result if this course be honestly pursued.

Calculations, Notes, and Records. Accurate calculations are the basis of correct proportions of machine parts. There is a right way to make calculations and a wrong way, and the student will usually take the wrong way unless he is cautioned at the start.

The wrong way of making calculations is the loose and shiftless fashion of scratching upon a scrap of detached paper marks and figures, arranged in haphazard form, and disconnected and incomplete. These calculations are in a few moments' time totally meaningless, even to the author of them himself, and are so easily lost or mislaid that when wanted they usually cannot be found.

Engineering calculations should always be made systematically, neatly, and in perfectly legible form, in some permanently bound blank book, so that reference may always be had to them at any future time for the purpose of checking or reviewing. Put all the data down. Do not leave in doubt the exact conditions under which the calculations were made. Note the date of calculation.

If a mistake in figures is made, or a change is found necessary, never rub out the figures or tear out the leaf, or in any way obliterate the figures. Simply draw a bold cross through the wrong part and begin again. Often a calculation which is supposed to be wrong is later shown to be right, or the facts which caused the error may be needed for investigation and comparison. Time which is spent in making figures is always valuable time, time too precious to be thrown away by destroying the record.

The recording of calculations in a permanent form, as just described, is the general practice in all modern engineering offices. This plan has been established purely as a business policy. In case of error it locates responsibility and settles dispute. Consistent designing is made possible through the records of past designs. Proposals, estimates, and bids may often be made instantly, on the basis of what these record books show of sizes and weights. This bookkeeping of calculations is as important a factor of systematic engineering as bookkeeping of business accounts is of financial success.

The student should procure for this purpose a good blank book with a firm binding, size of page not smaller than 6 by 8 inches (perhaps 8 by 11 inches may be better), and every calculation, however small and apparently unimportant, should be made in it.

Sample pages of engineering calculations are reproduced in Figs. 3 to 9. Note the sketch showing the forces. Note the clear statement of data. Note the systematic writing of the equations, and the definite substitutions therein. Note the heavy double underscoring of the result, when obtained. There is nothing in the whole process of the calculation that cannot be reviewed at any moment by anybody, and in the briefest time.

The development of a personal note-book is of great value to the designer of machinery. The facts of observation and experience recorded in proper form, bearing the imprint of intimate personal contact with the points recorded, cannot be equalled in value by those of any hand or reference book made by another. There is always a flavor about a personal note-book, a sort of guarantee, which makes the use of it by its author definite and sure.

The habit of taking and recording notes, or even knowing what notes to take, is an art in itself, and the student should begin early to make his note-book. Aside from the value of the notes themselves as a part of his personal equipment, the facility with which his eye will be trained to see and record mechanical things will be of great value in all of his study and work. How many men go through a shop and really see nothing of the operations going on therein, or, seeing them, remember nothing! An engineer, trained in this respect, will to a surprising degree be able to retain and sketch little details which fall under his eye for a brief moment only, while he is passing through a crowded shop.

Some draftsmen have the habit of copying all the standard tables of the various offices in which they work. While these are of some value in a few cases, yet this is not what is meant by a good note-book in the best sense. Ideas make a good note-book, not a mere tabulation of figures. If the basis upon which standards are founded can be transferred to permanent personal record, or novel methods of calculation, or simple features of construction, or data of mechanical tests, or efficient arrangement of machinery—if *these* can be preserved for reference, the note-book will be of greatest value.

Whatever is noted down, make clear and intelligible, illustrating by a sketch if possible. Make the note so clear that reference to it after a long space of years would bring the whole subject before the mind in an instant. If this is not done the author of the note himself will not have patience to dig out the meaning when it is needed; and the note will be of no value.

METHOD OF DESIGN.

The fundamental lines of thought and action which every designer follows in the solution of any problem in any class of work whatsoever, are four in number. The expert may carry all these in mind at the same time, without definite separation into a a step-by-step process; but the student must master them in their proper sequence, and thoroughly understand their application. In these four are concentrated the entire art of Machine Design. When they have become so familiar as to be instinctively applied on any and all occasions, good design is the result. The only other quality which will facilitate still further the design of good machinery is experience; and that cannot be taught, it must be acquired by actual work.

1. Analysis of Conditions and Forces. First, take a good square look at the problem to be solved. Study it from all sides, view it in all lights, note the worst conditions which can possibly exist, note the average conditions of service, note any special or irregular service likely to be called for.

With these conditions well in mind, make a careful analysis of all the forces, maximum as well as average, which may be brought into play. Make a rough sketch of the piece under consideration, and put in these forces. Be sure that these forces are at least approximately right. Go over the analysis carefully again and again. Remember that time saved at the beginning by hasty and poor analysis will actually be time lost at the end; and if the machine actually fails from this reason, heavy financial loss in material and labor will occur. Any haste toward completion of the structure beyond the roughest outline, without this careful study of forces, is a blind leap in the dark, entirely unscientific, and almost certain to result in ultimate failure.

On the other hand this principle may be carried too far. In trying to make the analysis thorough and the forces accurate, it is quite possible to consume more than a reasonable amount of time. Again, it is not always easy, and frequently impossible, to determine exactly the forces acting on a given piece. But their *nature*, whether sudden or slowly applied, rapid in action or only occurring at intervals, and their *approximate* direction and magnitude at least, are always capable of analysis. There are few, if any, cases where close assumptions cannot be made on the above basis and the design proceeded with accordingly. Hence the danger of too great refinement of analysis is simply to be avoided by the designer's plain business sense.

The first tendency of the student is to pass over the study of the forces as dull and dry, and attempt the design at once. He soon finds himself facing problems of which he sees no possible solution, and he bases his design on pure guess-work. This is the only solution possible from such a point of view, and is really no solution at all. A guess which has some rational backing is often successful; but in that case some analysis is required, and it is not a pure guess, but falls under the very principle we are considering.

There is no short cut to the design of machine parts which avoids this full understanding of the forces that they must sustain. The size of a belt depends upon the maximum pull upon it, and the designing of belts is nothing but providing sufficient cross-section of leather to prevent the belt tearing under the pull. Again, if pulley arms are not to break, or shafts twist off, or bolts be torn apart, or the teeth of gears fail, or keys and pins shear off, we must first, of course, find out what forces exist which are likely to produce stress that may lead to such breakage. We should not guess at the sizes, and then run the machine to see if breakage results, and then guess again. Machines are sometimes built in this way, but it is an unreasonable and uncertain method. We must use every effort to foresee the stress which a piece is liable to receive, before we decide its size. We must know all the forces approximately, if not positively. The analysis must be thorough enough to permit of reasonable assumption, if not positive assertion. It is manifestly impossible to solve any problem until we know exactly what the problem is; and a full analysis *is* the statement of the problem.

2. **Theoretical Design.** After we know by careful analysis what stress the machine part has to sustain, the next step is so to design it that it will theoretically resist the applied forces with the least expenditure of material.

We often see machinery with the metal of which it is made distributed in the worst possible manner. In places where the stress is heavy and a rigid member is needed, we find a weak, springy part; while in other parts, where there are no forces to be resisted, or vibration to be absorbed, there seems to be a waste of good material. Whether in such case the analysis of the forces was poor, or perhaps not made at all, or whether a knowledge of how to design so as to resist the given forces was wholly absent, cannot be told. At any rate, lack of either or both is clearly shown in the result.

Any member of a machine may vary in form from a solid block or chunk of material to an open ribbed structure. The solid chunk fills the requirement as far as strength is concerned, unless it is so heavy as to fail from its own weight. But such construction is poor design, except in cases where the concentration of heavy mass is necessary to absorb repeated blows like those of a hammer. The possibility of these blows should, however, have been determined in the analysis; and the solid, anvil construction then becomes theoretical design for that analysis.

For steadily applied loads an open, ribbed, or hollow box structure can be made which will distribute the metal where it is theoretically needed, and each fiber will then sustain its proper share of the load. In this way weight, cost, and appearance are heeded; and the service of the piece is as good as, and probably better than, it would be with the clumsy, solid form.

There is no such thing as putting too much theory into the design of machinery. The strongest trait which an engineer can have is absolute faith in his analysis and calculations, and their reproduction in his theoretical design. Theoretical design is an indication of scientific advance in the art, and some of the greatest steps of progress which have been made in recent years have been accomplished through a purely theoretical study of machine structure.

It will never do, however, to be satisfied with theoretical design when it is not in accord with modern commercial and manufacturing considerations. Hence the next step after the determination of the theoretical design is the study of it from the producing standpoint.

3. Practical Modification. All theoretical design viewed from the business standpoint is worthless, unless it has been subjected to the test of cheap and efficient production. Each machine detail, though correct in theory, may yet be improperly shaped and unfit for the part it is to play in the general scheme of manufacture.

The conditions here involved are changeable. What is good design in this decade may be bad in the next. In this light the designer must be a close student of the signs of the times; he must follow the march of progress, closely applying existing resources, conditions, and facilities, otherwise he cannot produce up-to-date designs. The introduction of new raw materials, the cheapening of production of others, the changing of shop methods, the use of special machinery, the opening of new markets, the development of new motive agents,—all these and many others are constantly demanding some modification in design to meet competition.

Illustrative of this, note the change which has been wrought by the development of electric power, the rise and decline of the bicycle business, the present manufacture of automobiles, the last named especially with reference to the development of the small motive unit, the gasolene engine, the steam engine, etc. The design of much machinery has been materially changed to meet the exacting demands of these new enterprises.

Practical modifications of design necessary to meet the limitations of construction in the pattern shop, foundry, and machine shop are of daily application in the designer's work. He must keep in his mind's eye at all times the workmen and the processes they use to create his designs in metal in the shop.

"How can this be made?" "Can it be made at all?" "Can it be made cheaply?" "Will it be simple in operation after it is made?" "Can it be readily removed for repair?" "Can it be lubricated?" "How can it be put in place?" "How can it be gotten out?" "Will it be made in small quantities or large?" "Will it sell as a special or standard machine?" etc., etc.

The consideration of such questions as these is a practical necessity as a business matter. No other feature affects the design of machinery more, perhaps; for designs which cannot be built as business propositions are no designs at all.

The student, it is true, may not have the extended shop knowledge which is essential to this; but he can do much for himself by visiting shops whenever possible, getting hold of shop ways of doing things, and invariably treating his work as a business matter. Though a man may not be a pattern maker, molder, blacksmith, or machinist, yet he can soon gain ideas of the processes in each of these branches which will be of immense advantage to him in his designing work.

4. **Delineation and Specification.** This means the clear and concise representation of the design by mechanical drawings.

This is as much a part of the routine method of Machine Design as the other three points which have been discussed. The mere act of putting the results of mechanical thinking on paper is one of the greatest helps to force thinking machinery to systematic and definite action. A designer never thinks very long without drawing something, and the student must bring himself to feel that a drawing in its first sense is a means of helping his own thought, and must freely use it as such.

In its second and final sense, the drawing is an order and specification sheet from the designer to the workman. Design which stops short of exact, finished delineation in the form of working shop drawings is only half done. In fact the possibility of a piece being thus exactly drawn is often the crucial test of its feasibility as a part of a machine. It is easy to make general outlines, but it is not so easy to get down to finished detail. It is safe to say that there is no one thing productive of more trouble, delay and embarrassment, and waste of time and money in the shop, when there need be none from this cause, than a poor detail drawing. The efficiency of the process of design is not fully realized, and failures are often recorded where there should be success, merely because the indefiniteness permitted by the designer in the drawings naturally transmitted itself to the workman, and he in turn produced a part indefinite in form and operation.

The actual process of drawing in the development of a design may be outlined as follows :

Rough sketches merely representing ideas, not drawn to scale, are first made. These are of use only so far as the choice of mechanical ideas is concerned, and to carry preliminary dimensions.

Following these sketches, comes a layout to scale, of the favored sketch, a working out of the relative sizes and location of the parts. This drawing may be of a sketchy nature, carrying a principal dimension here and there to fix and control the detailed design. In this drawing the design is developed and general detail worked out. The minute detail of the individual parts is, however, left to the subsequent working drawing.

This layout drawing may now be turned over to an expert draftsman or detail designer, who picks out each part, makes an exact drawing of it, studying every little detail of its shape, and finally adds complete dimensions and specifications so that the workman is positively informed as to every point of its construction.

General drawings and cross sections constitute the last step in the process of complete delineation. These show the parts assembled in the complete machine. They also serve a valuable purpose to the draftsman in checking up the dimensions of the detail drawings. Errors which have escaped previous notice are often discovered in this way. The layout, mentioned above, is sometimes finished up into a general drawing; but it is safer to make an entirely new drawing, as changes in detail are often necessary after the layout is made.

The four fundamental lines of thought and action noted above may be summarized thus—"**analyze and theorize, modify and delineate.**" This is a maxim easy to remember, applicable to every problem in Machine Design, and always provides the answer to the question "What shall I do, how shall I proceed?" by pointing out the proper sequence in the course to be followed.

CONSTRUCTIVE MECHANICS.

Mechanics is a constructive science, its principles lying at the root of the design and operation of all machinery. It is usually taught, however, as an advanced mathematical subject; and the student gets his original conceptions of forces, moments, and beams in the abstract, before he realizes the constructive value of such conceptions. By "Constructive Mechanics" is meant the study of a machine purely from its constructive side, the viewing of the parts with respect to their "mechanics," and satisfying the requirements of the same in form and arrangement.

The student may cultivate this habit of clear, mechanical perception by constantly noting the "mechanics" of the simple structures which he sees in his daily routine of work. Aside from machinery, in which the "mechanics" is often obscure, the world is full of simple examples of natural strength and symmetry, explainable by application of the principles of pure "mechanics."

Posts and pillars are largest at their bases; overhanging brackets or arms are spread out at the fastening to the wall; heavy swinging gates are counter-balanced by a ponderous weight; the old-fashioned well sweep carries its tray of stones at the end, adjusting the balance to a nicety; these are examples of things depending for their form and operation upon the principles of "mechanics." The building of them involved "constructive mechanics," and yet their constructor perhaps never heard of the science, using merely his natural sense of mechanical fitness. Such simple reasoning is, however, Constructive Mechanics.

Forces, Moments, and Beams. Machines are nothing but a collection of (1) parts taking direct stress, or (2) parts acting as loaded beams. Forces acting *without* leverage produce direct stress on the sustaining part. Forces acting *with* leverage produce a moment; the sustaining member is a beam, and the stress therein depends on the theory of beams, as explained in "Mechanics."

An example of the first is the load on a rope, the force acting without leverage, and the rope therefore having a direct stress put upon it.

An example of the second is a push of the hand on the crank of a grindstone. A moment is produced about the hub of the crank; the arm of the crank is a beam, and the stress at any point of it may be found by the method of theory of beams.

Tension, Compression, and Torsion. The stress induced in the sustaining part, whether tensile, compressive, or torsional, is caused by the application of forces, either acting directly without leverage, or with leverage in the production of moments.

The forces applied from external sources are at constant war with the resisting forces due to the strength of the fibres of the material composing the machine members. The moments of the *external* forces are constantly exerted against and balanced by the moments of the *internal* resistance of the material. Hence, design, from a strength standpoint, is merely a balancing of internal strength against external force. In other words, we may in all cases write a sign of equality, place the applied effort on one side, the effective resistance on the other, and we shall have an equation, which, if capable of solution, will give the proper proportions of the parts considered.

External Force = Internal Resistance.

External Moment = Internal Moment of Resistance.

Expressed in terms of the "Mechanics:"

$$P=AS \qquad (1)$$

$$B \text{ or } T=\frac{SI}{c} \qquad (2)$$

In these formulas, which are perfectly general,

P=direct load in pounds.

A=area of effective material, in square inches.

S=working fibre stress of the material (tensile, compressive, or shearing), in pounds per square inch.

B or T=external moment (bending or torsional), in inch-pounds.

I=moment of inertia (direct or polar), of the resisting section.

c=distance of the most remote fibre of the resisting section from the neutral axis.

P may produce direct tensile, compressive, or shearing stress.

B may produce tensile or compressive stress, and requires use of direct moment of inertia in either case.

T produces shearing stress, and requires use of polar moment of inertia.

The origin of formula (1) is obvious, the assumption being that the fibre stress is equally distributed to every particle in the area "A."

The development of formula (2) is given in any text-book in Mechanics. It requires the aid of the Calculus, however. Any good handbook gives values for both the direct moment of inertia and the polar moment of inertia for quite a large variety of sections, so that further reference is an easy matter for the student. These values are also obtained through the methods of the Calculus.

The reason for introducing these formulas at this time is to call the attention of the student especially to the fact of their universal and fundamental use in all problems concerning the strength of machine parts. Nearly every computation may be reduced to or expanded from these two simple equations. Many complex combinations occur, of course, which will not permit simple and direct application of these formulas, but the student will do well to place himself in perfect command of these two. Assuming that he is able to analyze forces, and compute the simple moment at the point where he wishes to find the strength of section, the rest is the mere insertion of the assumed working fibre stress of the material in the formula (2) above, and solution for the quantity desired.

When the case is one of combined stress, the relation becomes more complicated and difficult of analysis and solution. The most common case is where bending is combined with torsion, as in the case of a shaft transmitting power, and at the same time loaded transversely between bearings. In fact there are very few cases of shafts in machines, which, at some part of their length, do not have this combined stress. In this case the method of procedure is to find the simple bending moment and the simple torsional moment separately, in the ordinary way. Then the theory of elasticity furnishes us with a formula for an equivalent bending or an equivalent torsional moment which is supposed to produce the same effect upon the fibres of the material as the combined action of the two simple moments acting together. In other words, the separate moments combined in action, being impossible of solution in that form, are reduced to an equivalent simple moment and the solution then becomes the same as for the previous case.

These equivalent equations are given below, the subscript "e" being added to express separation from the simple moment:

$$B_e=\frac{B}{2}+\tfrac{1}{2}\sqrt{B^2+T^2} \qquad (3)$$

$$T_e=B+\sqrt{B^2+T^2} \qquad (4)$$

B_e and T_c, found from these equations, are the external moments, and are to be equated to the internal moments of resistance of the section precisely as if they were simple bending or torsional moments. Either may be used. For shafts (4) is generally used, being the simpler of the two in form.

FRICTION AND LUBRICATION.

The parts of a machine which have no relative motion with regard to each other are not dependent upon lubrication of their surfaces for the proper performance of their functions. In cases where relative motion does occur, as between a planer bed and its ways, a shaft and its bearing, or a driving screw and its nut, friction, and consequent resistance to motion, will inevitably occur. Heat will be generated, and cutting or scoring of the surfaces will take place if the surfaces are allowed to run together dry.

This difficulty, which exists with all materials, cannot be overcome, for it is a result of roughness of surface, characteristic of the material even when highly finished. The problem of the designer, then, is to take conditions as he finds them, and, as he cannot change the physical characteristics of materials, so choose those which are to rub together in the operation of the machine that friction will be reduced to the lowest possible limit. Now it fortunately happens that there are certain agents like oil and graphite, which seem to fill up the hollows in the surface of a solid material, and which themselves have very little friction on other substances. Hence, if a machine permits by its design an automatic supply of these lubricating agents to all surfaces having motion between them, friction may be reduced to the lowest limit.

If this full supply of lubricant be secured, and the parts still heat and cut, then the fault may be traced to other causes, such as springy surfaces, localization of pressure, or insufficient radiating surface to carry away the heat of friction as fast as it is generated.

Lubricating agents are of a nature running from the solid graphite form to a thick grease, then to a heavy dark oil, and finally to a thin, fluid oil flowing as freely as water. The solid and heavy lubricants are applicable to heavily loaded places where the pressure would squeeze out the lighter oils. Grease, forced between the surfaces by compression grease cups, is an admirable lubricator for heavy machinery under severe service. High-speed and accurate machinery, lightly loaded, requires a thin oil, as the fits would not allow room for the heavier lubricants to find their way to the desired spot. The ideal condition in any case is to have a film of lubricant always between the surfaces in contact, and it is this condition at which the designer is always aiming in his lubricating devices.

Oil ways and channels should be direct, ample in size. readily accessible for cleaning, and distributing the oil by natural flow over the full extent of the surface. Hidden and remote bearings must be reached by pipes, the mouths of which should be clearly indicated and accessible to the operator of the machine. Such pipes must be straight, if possible, and readily cleaned.

There is one practical principle affecting the design of methods of lubrication of a machine which should be borne in mind. This is, "Neglect and carelessness by the operator *must* be provided for." It is of no use to say that the ruination of a surface or hidden bearing is due to neglect by the operator, if the means for such lubrication are not perfectly obvious. This is "locking the door after the horse is stolen." The designer has not done his duty until he has made the scheme of lubrication so plain that every part *must* receive its proper supply of oil, except by gross and willful negligence, for which there can be no possible just excuse.

WORKING STRESSES AND STRAINS.

Some persons object to the use of these terms, as one is frequently used for the other, and misunderstanding results. This is doubtless true; but the student may as well learn the true relation of the terms once for all, because he will frequently run across them in his reading and reference work, and should interpret them rightly. The strict relation of the two is as follows:

Stress is the internal force in a piece resisting the external force applied to it. A weight of ten pounds hanging on a rope produces a *stress* of ten pounds in the rope.

Strain is the change of shape, or deformation, in a piece resisting an external force applied to it. If the above weight of ten pounds stretches the rope $\frac{1}{4}$ inch, the *strain* is $\frac{1}{4}$ inch.

Unit stress is stress per unit area, e. g., per square inch.

Unit strain is strain per unit length, e. g., per inch length.

In the above case, if the rope were $\frac{1}{2}$ square inch in area and 30 inches long, the unit stress, or intensity of stress, is $10 \div \frac{1}{2} = 20$ pounds per square inch; the unit strain is $\frac{1}{4} \div 30 = \frac{1}{120}$ inch per inch.

When stress is induced in a piece, the strain is practically proportional to the stress for all values of the stress below the elastic limit of the material; and when the external load is removed the strain will entirely disappear, or the recovering power of the material will restore the piece to the original length.

Illustrating by the case above, on the supposition that the elastic limit has not been reached by the stress of 20 pounds per square inch, if the load of 10 pounds were taken off, the $\frac{1}{4}$-inch strain would disappear and the rope return to its original length; if the load were changed to $\frac{1}{2}$ of 10 pounds, or 5 pounds, the strain would be $\frac{1}{2}$ of $\frac{1}{4}$ inch, or $\frac{1}{8}$ inch.

Now it is found that if we wish a piece to last in service for a long time without danger of breakage, we must not permit it to be stressed anywhere near the elastic limit value. If we do, although it will probably not break at once, it is in a dangerous condition, and not well suited to its requirements as a machine member. The technical name for this weakening effect is "fatigue." It is further found that the fatigue due to this repeated stress is reached at a lower limit when the stress is alternating in character than when it is not. In other words, if we first pull on a piece and then push on it, we shall first have the piece in tension and then in compression; this alternation of stress repeated to near the elastic limit of the material will fatigue it, or wear out the fibres, and it will finally fail. If, however, we first pull on the piece with the same force as before, and then let go, we shall first have the piece in tension and then entirely relieved; such repetition of stress will finally "fatigue" the material, but not so quickly as in the first case. Experiments indicate that it may take twice as many applications in the latter case as in the former.

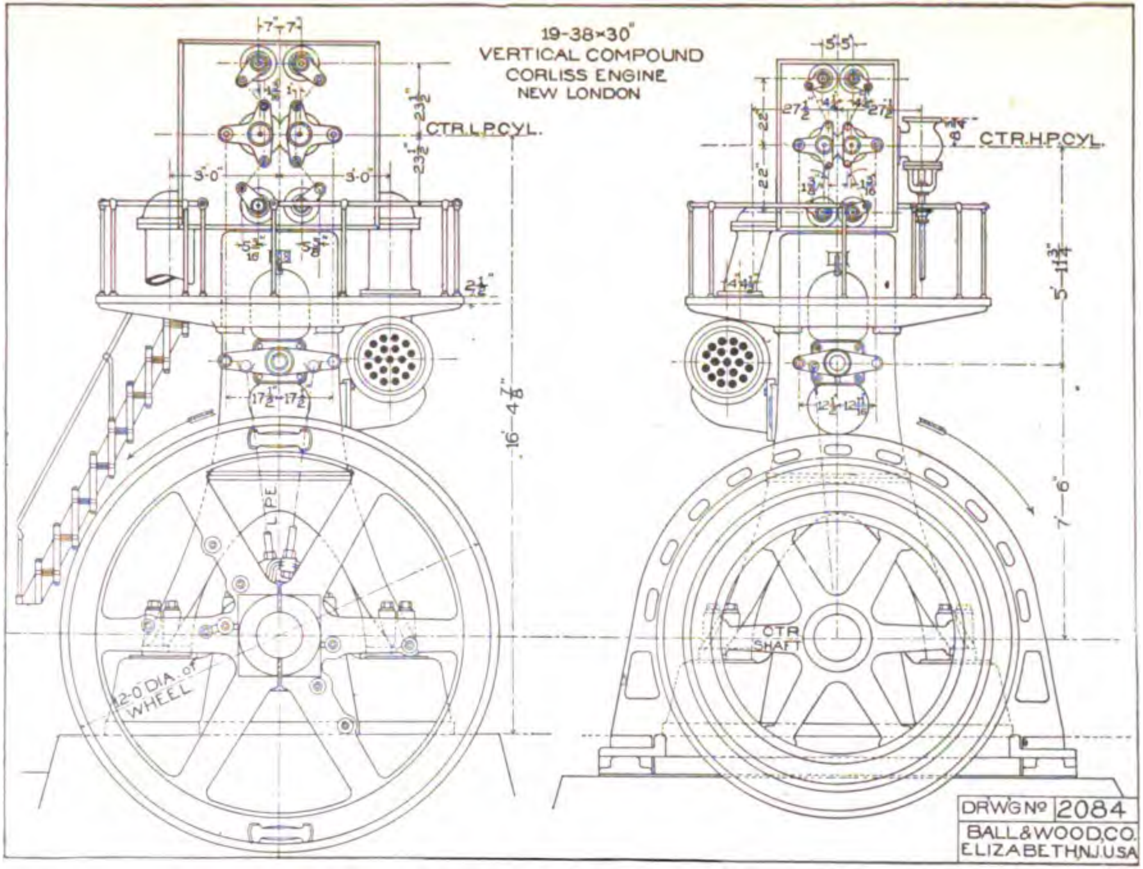
The working stress of materials permissible in machines is based on the above facts. The breaking strength divided by a liberal factor of safety will not necessarily give a desirable working stress. The question to be answered is, "Will the assumed working fibre stress permit an indefinite number of applications of the load without fatiguing the material?"

Hence we see that the same material may be safely used under different assumptions of working stress. For example, a rotating shaft, heavily loaded between bearings. acts as a beam which in each revolution is having its particles subjected, first to a maximum tensile stress, and then to a maximum compressive stress. This is obviously a very different stress from that which the same piece would receive if it were a pin in a bridge truss. In the former we have a case where the stress on each particle reverses at each revolution, while in the latter we have merely the same stress recurring at intervals, but never becoming of the opposite character. For ordinary steel, a value of 8,000 would be reasonable in the former case, while in the latter it may be much higher with safety, perhaps nearly double.

From the facts stated above, it is evident that exact values for working fibre stress cannot be assumed with certainty and applied broadly in all cases. If the elastic limit of the material is definitely known we can base our working value quite surely on that.

With but a general knowledge of the elastic limit, ordinary steel is good for from 12,000 to 15,000 pounds per square inch non-reversing stress, and 8,000 to 10,000 reversing stress. Cast iron is such an uncertain metal on account of its variable structure that stresses are always kept low, say from 3,000 to 4,000 for non-reversing stress, and 1,500 to 2,500 for reversing stress.

With these values as a guide, and the special conditions controlling each case carefully studied, reasonable limits may be



DRAWING OF VERTICAL COMPOUND CORLISS ENGINE.

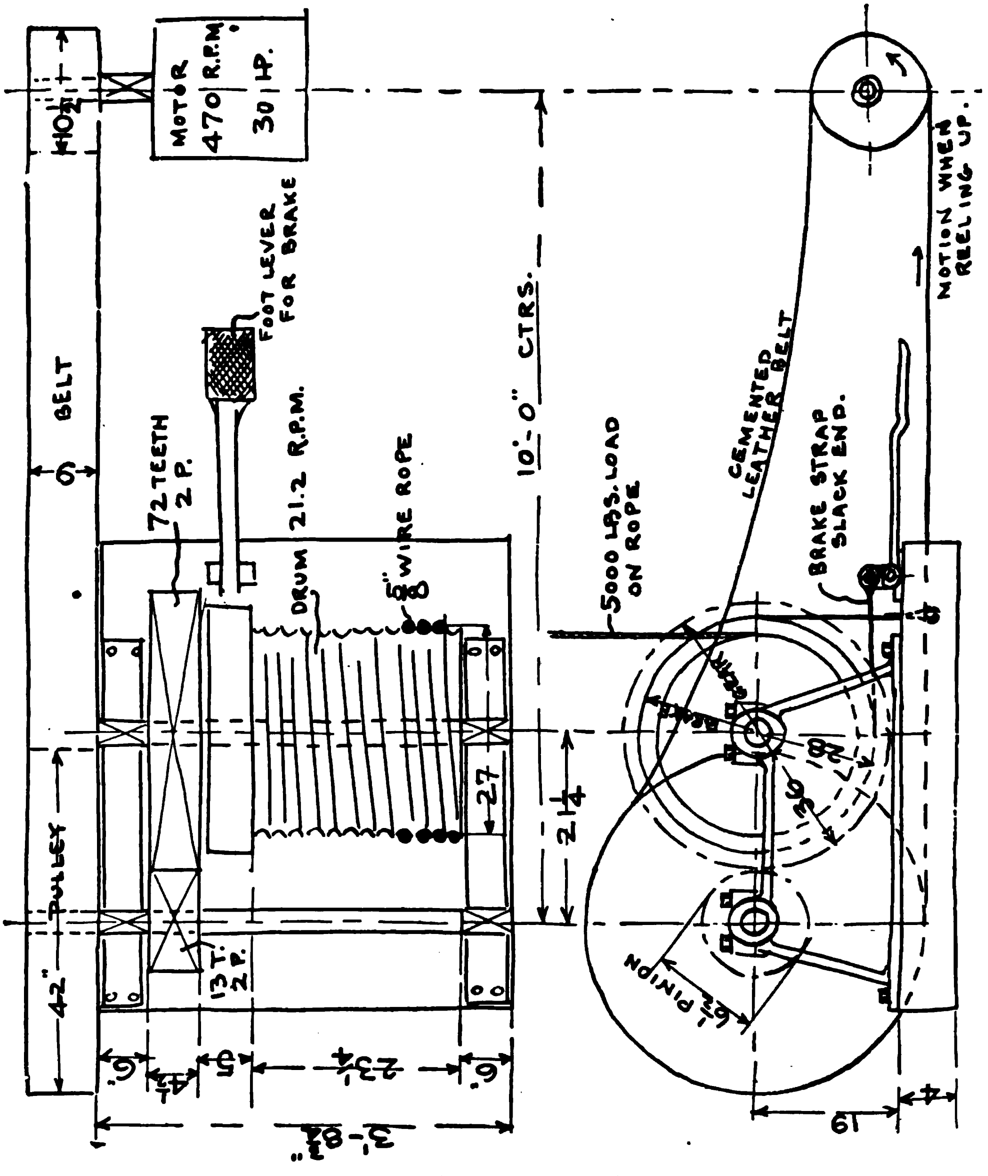
Reproduced from a blue print.

assigned for working stress, not only of steels, various grades of cast iron, and mixtures of the same, but of other alloys, brass, bronze, etc. Gun metal, semi-steel, and bronze are intermediate in strength between cast iron and steel. Data on the strength of materials are available in any of the handbooks, and should be consulted freely by the student. They will be found somewhat conflicting, but will assist the judgment in coming to a conclusion.

Application to Practical Case. In actual practice the only information which the designer has, upon which to base his design, is the object to be accomplished. He must choose or originate suitable devices, develop the arrangement of the parts, make his own assumptions regarding the operation of the machine, then *Analyze and Theorize, Modify and Delineate* each detail as he meets it.

This, it will be found, is a very different matter from taking some familiar piece of machinery, such as a pulley, or a shaft, or a gear, as an isolated case, the load being definitely given, and proceeding with the design. This is easily done, but is only half the problem, for machine parts, such as pulleys, gears, and shafts, do not confront the designer tagged or labeled with the conditions they are to meet. He is to provide parts to meet the specific conditions, and it is as much a part of his designing method to know how to attack the design of a machine as it is to know how to design the parts in detail after the attack has reduced the members to definitely loaded structures. The whole process must be gone through, the preliminary sketches, calculations, and layout, all of which precede the detail design and working drawings; and no step of the process can be omitted.

It is for this reason that the present case used for illustration is carried out quite thoroughly. The student should make himself familiar with every step of the designing method as applied to this simple case of design. More complex problems, handled in the same way, will simplify themselves; and when the point is reached where confidence exists to take hold of the design of any machine, however unfamiliar its object may be, or however involved its probable detail appears, the student has become the true designer. It is the knowing how to attack a problem, to start definite work on it, to go ahead boldly, confident that the method applied will produce results, that gives command of the design of machinery and wins engineering success.



The special case which has been chosen to illustrate the application of the principles stated in the foregoing pages is ideal, in that it does not represent any actual machine at present in operation. Probably builders of hoisting machinery have devices which would improve the machine as shown. In detail, as well as arrangement, they could doubtless make criticism as manufacturers. The arrangement as shown is merely intended to bring out in simplest form the common elements of transmission machinery as parts of some definite machine, instead of as isolated details. The design is one entirely possible, practical, and mechanical, but special attention has been paid to simplicity in order to enable the student to follow the method closely, for the *method* is the chief thing for him to acquire.

The student is expected to refer constantly to Part II for a more formal and general discussion of the simple machine elements involved in the case considered. Part II is intended to be a simplified and condensed reference book, carried out in accordance with the method of machine design as specified in Part I. The student should not wait until he has completed the study of this part before taking up Part II, for the latter is intended for use with the former in the solution of the problems.

In the case of power transmission about to be studied, the running, conversational method employed assumes that the student is in possession of the matter in Part II on the subject considered. Thus, in the design of the pulley, reference to the subject of "Pulleys" in Part II is necessary to follow the train of calculation; in designing the gear, consult "Gears;" in calculating size of shafts, see "Shafts," etc., etc.

Problem. A machine is to be designed to be set on the floor of a building to drive a wire rope falling from the overhead sheaves of an elevator or hoist. Without regard to details of this overhead arrangement, for its design would be a separate problem, suppose that the data for the rope are as follows:

Load on rope......................	5,000 pounds.
Speed of rope......................	150 feet per minute.
Length of rope to be reeled in......	200 feet.

We shall further assume that the driving power is to be an electric motor belted to the machine, that the required speed reduction can be satisfactorily obtained by a single pair of pulleys and one pair of gears, and that a plain band brake is to be applied to the drum.

With this data we shall proceed to work out the detail design of the machine.

Preliminary Sketch. The first thing to do is to sketch roughly the proposed arrangement of the machine.

This might appear like Fig. 1 except that it would have no dimensions in addition to the data given above. If the scheme seems suitable, the next step is to make such preliminary calculations as will give further data, exact or closely approximate sizes, to be put at once on the sketch, to outline the future design.

Rope and Drum. Referring to tables of strength of wire rope (Kent's Pocket Book gives the manufacturers' list), we find that a $\frac{5}{8}$-inch cast-steel rope will carry 5,000 pounds safely, and that the proper size of drum to avoid excessive bending of the rope around it is 27 inches diameter.

Allowing $\frac{1}{8}$ inch between the coils as the rope winds on the drum, the pitch of coil will be $\frac{3}{4}$ inch as shown in sketch, Fig. 2. The length of one complete coil is, practically, $\frac{27 \times 3.1416}{12} = 7.07$ feet. To provide for 200 feet will require $\frac{200}{7.07} = 28+$coils. To be safe, let us provide for 30 coils, for which a length of drum $(30 \times \frac{3}{4}) + \frac{3}{4} = 23\frac{1}{4}$ inches is required.

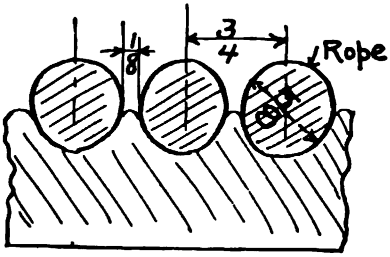


Fig. 2.

The space for brake strap may be assumed at 5 inches, and the thickness to provide necessary strength determined later in the design. The frictional surface of the strap may be of basswood blocks, say $1\frac{1}{4}$ inches thick, screwed to the metal band. The diameter of brake surface may be 28 inches.

Driving Gears. The size of drum gear evidently depends upon the method of fastening to the drum, and, other things being equal, should be kept as small as possible. One way would be to key the gear on the outside of the drum, another to bolt the gear to the end of the drum. The latter has the advantage that a standard gear pattern can be used with the slight change of addition of bolt flange on the arms. This makes a simple, direct, and strong drive, the bolts being in shear.

Sketching this arrangement as the preferred one (Fig. 2A), it is evident that the diameter of the gear should be at least as large as the drum in order to keep the tooth load down to a reasonable figure. On the other hand, if made too large, it spreads out the machine and destroys its compactness. As a diameter of 36 inches is not excessive, let us assume this, and see if a desirable proportion of gear tooth can be found to carry the load.

For a pitch diameter of 36 inches there will be a theoretical load of $\frac{5{,}000 \times 27}{36} = 3{,}750$ pounds at the pitch line. But the load on the tooth must not only impart a pull of 5,000 pounds to the rope, but must overcome friction between the gear teeth in action, also between the drum shaft and its bearings. Assuming the efficiency between the rope and tooth load to be 95 per cent, the net load, therefore, which the tooth must take is $\frac{3{,}750}{.95} = 3{,}947$, say 4,000 pounds.

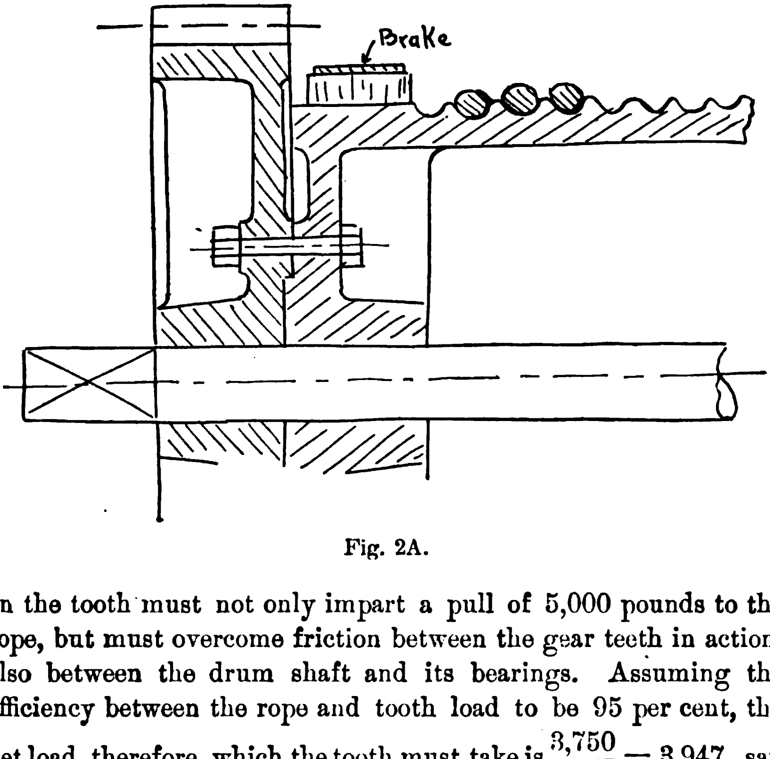


Fig. 2A.

Assuming involute teeth, and applying the "Lewis" formula (Part II, "Gears"):

$$W = s \times p \times f \times y$$

$$4{,}000 = 6{,}000 \times p \times f \times .116$$

$$p \times f = \frac{4{,}000}{6{,}000 \times .116} = 5.7 \text{ inches}$$

$W = 4{,}000$
$s = 6{,}000$
$y = .116$ (number of teeth assumed at 75)
p = circular pitch
f = face of gear

Let $f = 3p$ (a reasonable proportion for machine-cut teeth).

Then $3 \times p^2 = 5.7$
$p^2 = 1.9$
$p = \sqrt{1.9} = 1.378$ inches

The diametral pitch corresponding to this is

$$\frac{3.1416}{1.378} = 2.28$$

which is just between the regular standard pitches, 2 and 2½, for which stock cutters are made. To be safe, let us take the coarser pitch, which is 2. The circular pitch corresponding to this is $\frac{3.1416}{2} = 1.57$, and making the face about three times the circular pitch gives

$$3 \times 1.57 = 4.71, \text{ say } 4\tfrac{1}{2} \text{ inches.}$$

The number of teeth in the gear is then $36 \times 2 = 72$. Referring to the value assumed for the tooth factor in calculation above, it is seen that y was based on 75 as the number of teeth, which is near enough to 72 to avoid the necessity of further checking the result.

The pinion to mesh with this gear should be as small as possible in order to get a high-speed ratio between pinion shaft and drum, otherwise an excessive ratio will be required in the pulleys, making the large one of inconvenient size. Small pinions have the teeth badly undercut and therefore weak, 13 teeth being the lowest limit usually considered desirable, for that reason. Choosing that number, we have a pitch diameter of $\frac{13}{2} = 6.5$ in., which is probably ample to take the shaft and key, and still leave sufficient stock under the tooth for strength. If made of cast iron, however, the pinion teeth, on account of the low number, will be narrower at the root than those of the gear of 72 teeth. Yet it was upon the basis of the latter that the pitch was chosen, for it will be remembered that the value of y in the formula was taken at .116. Hence the pinion will be weaker than the gear unless we make it of stronger material than cast iron, of which the large gear is supposed to be made. Steel lends itself very readily to this requirement; and in practice, pinions of less than 20 teeth are usually made of this material, hence we shall specify the pinion to be of steel.

Pulleys. The question now is whether or not we can get a suitable ratio in the pulleys without making the large one of inconvenient size, or giving the motor too slow speed for an economical proportion.

Suppose we limit ourselves to a diameter of 42 inches for the large pulley, and try a ratio of 4 to 1; this will give a diameter for the small pulley of $\frac{42}{4}=10\frac{1}{2}$ inches. We shall then have

Total ratio between drum and motor........ $\frac{72}{13}\times 4=\frac{288}{13}=22.2$

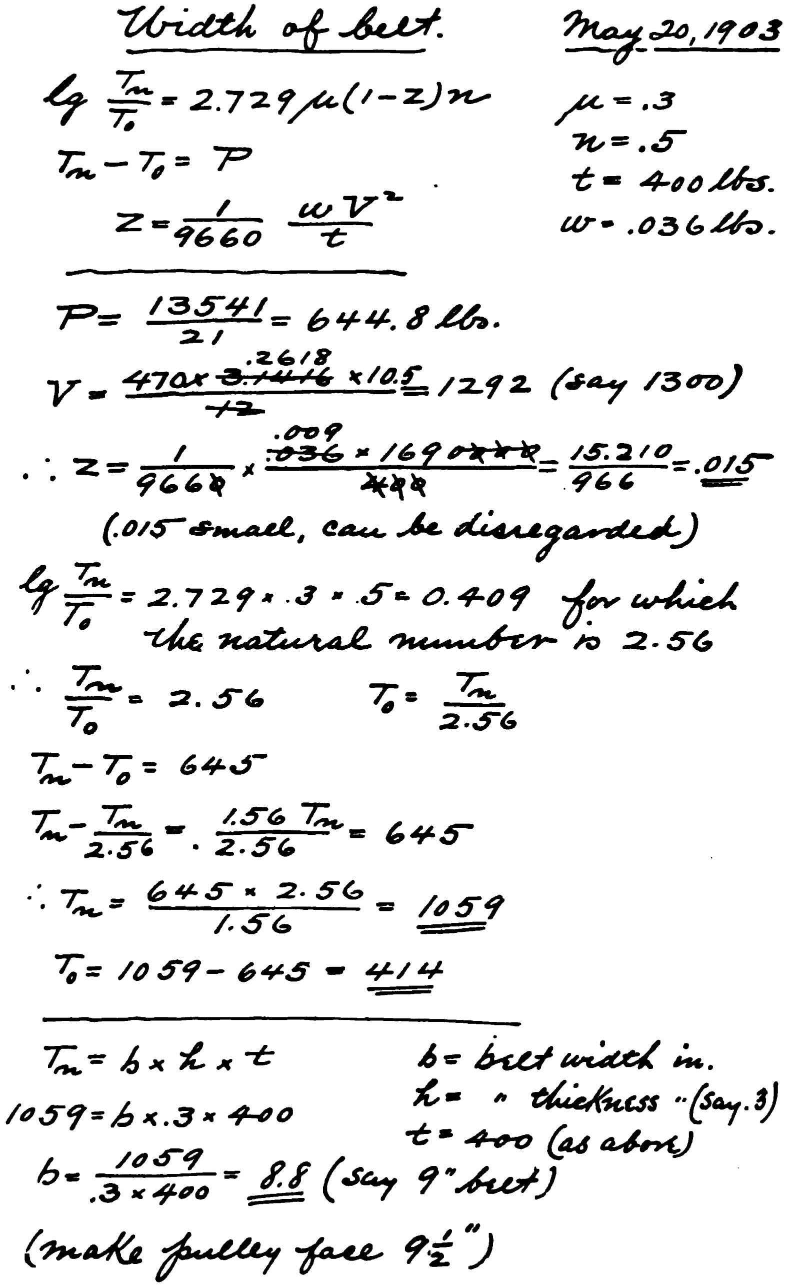
Rev. per min. of drum to give 150 f. p. m. of rope .... $\frac{150}{7.07}=21.2$

Rev. per min. of motor......................$22.2\times 21.2=470$

Horse-power of motor at 80 per cent efficiency $\frac{150\times 5{,}000}{33{,}000\times .80}=30$

A 30 H. P. motor running 470 r. p. m. would be classed as a slow speed motor and would be a heavier machine and cost more than one of higher speed. It will be noticed, however, that the diameter of the small pulley is already quite reduced, and it is hardly desirable to decrease it still further. Neither can we increase the large pulley, as we have already set the limit at 42 inches. Hence, for our present problem we cannot improve matters much without increasing the size of the large gear, which is undesirable, or putting in another pair of gears, which is contrary to the conditions of the problem. As such a motor is perfectly reasonable, we shall assume it to be chosen for the purpose.

In commercial practice it would be well to pick out some standard make of motor of the required horse-power, note the speed as specified by the makers, and then, if possible, suit the ratio in the machine to this speed. It is always best to use standard machinery, if possible, both from the standpoint of first cost, as well



Width of belt. May 20, 1903

$$\lg \frac{T_n}{T_0} = 2.729\mu(1-z)n \qquad \mu = .3$$

$$n = .5$$

$$T_n - T_0 = P \qquad t = 400 \text{ lbs.}$$

$$z = \frac{1}{9660}\,\frac{wV^2}{t} \qquad w = .036 \text{ lbs.}$$

$$P = \frac{13541}{21} = 644.8 \text{ lbs.}$$

$$V = \frac{470 \times \overset{.2618}{\cancel{3.1416}} \times 10.5}{\cancel{12}} = 1292 \text{ (say 1300)}$$

$$\therefore z = \frac{1}{966\cancel{0}} \times \frac{\overset{.009}{\cancel{.036}} \times 169\cancel{0000}}{\cancel{400}} = \frac{15.210}{966} = \underline{\underline{.015}}$$

(.015 small, can be disregarded)

$$\lg \frac{T_n}{T_0} = 2.729 \times .3 \times .5 = 0.409$$ for which the natural number is 2.56

$$\therefore \frac{T_n}{T_0} = 2.56 \qquad T_0 = \frac{T_n}{2.56}$$

$$T_n - T_0 = 645$$

$$T_n - \frac{T_n}{2.56} = \frac{1.56\,T_n}{2.56} = 645$$

$$\therefore T_n = \frac{645 \times 2.56}{1.56} = \underline{\underline{1059}}$$

$$T_0 = 1059 - 645 = \underline{\underline{414}}$$

$$T_n = b \times h \times t \qquad b = \text{belt width in.}$$

$$1059 = b \times .3 \times 400 \qquad h = \text{" thickness " (say .3)}$$

$$t = 400 \text{ (as above)}$$

$$b = \frac{1059}{.3 \times 400} = \underline{\underline{8.8}} \text{ (say 9" belt)}$$

(make pulley face $9\frac{1}{2}$")

Fig. 3

as ease of replacing worn parts. Machinery ordered special is expensive in first cost of designing, patterns, and tools, and extra spare parts for emergency orders are not often kept on hand.

Tabulation of Torsional Moments. For future reference, it is desirable at this point to tabulate the torsional moment, or torque, about each of the three shaft axes, assuming reasonable efficiencies for the various parts, as follows:

Efficiency between drum and gear tooth.............95 per cent
Efficiency between drum and pinion shaft...........90 per cent
Efficiency between drum and motor shaft ....80 per cent

TABLE OF TORSIONAL MOMENTS.

Axis.	Inch Lbs. Torque at 100 Per Cent Efficiency.	Inch Lbs. Torque, Efficiency as Above.
Drum	$5{,}000\times\frac{27}{2}$ ....$=67{,}500$	$\frac{67{,}500}{.95}=71{,}052$
Pinion......	$5{,}000\times\frac{27}{2}\times\frac{13}{72}$ ......$=12{,}187$	$\frac{12{,}187}{.90}=13{,}541$
Motor	$5{,}000\times\frac{27}{2}\times\frac{13}{72}\times\frac{10.5}{42}=3{,}047$	$\frac{3{,}047}{.80}=3{,}809$

This means that the motor develops a torque of 3,809 inch-pounds delivering to pinion shaft 13,541 inch-pounds, and to drum 71,052 inch-pounds.

Width of Belt. The page of calculation for belt width is reproduced in Fig. 3.

The calculation as given is strictly scientific, based on the working strength of a cemented joint ($t=400$ lbs. per square inch). This is a favorable situation for the use of a cemented joint, because it is easy to provide means of adjusting the belt tension by placing the motor on a sliding base. Otherwise a laced joint could be used, requiring relacing when the belt slackens through its stretch in service. Under the assumption that a double *laced* belt is used, the empirical formula below is one often applied:

$$\text{H. P.}=\frac{w\times V}{540}=\frac{w\times 1{,}300}{540}=30$$

This gives $w=\frac{540\times 30}{1{,}300}=12{\cdot}4$ inches (say 12 inches).

It should be remembered that this value is purely empirical; it applies to a *laced* joint, and could not be expected to check the value of 9 inches obtained by the first computation for a cemented joint. It is fairly in proportion. For the quite definite service required of the belt in the present case, the width of 9 inches is doubtless sufficient, considering the cemented joint.

Length of Bearings. Considerable latitude in choice of length of bearings is permissible, especially in such slow-speed machinery. There is probably little danger from heating, and the question then becomes one of wear. It is better in such cases as the one in question, to choose boldly a length which seems to be reasonable and proceed with the design on that basis, even if the length be later found out of proportion to the shaft diameter, than to waste too much time in the preliminary calculation over the exact determination of this question. Probably in most cases of commercial practice the existence of patterns, or some other practical consideration, will decide the limits of length.

In the present instance it seems reasonable that a length of 6 inches would fill the requirement for the worst case, that of the drum shaft, and it is obvious that the bearings for the pinion shaft would naturally be of the same length on account of being cast on the same bracket, and faced at the same setting of the planer tool.

Height of Centers. The large pulley should naturally swing clear of the floor. This will require, say, a total height of 23 inches, out which we may take 4 inches for the base, leaving 19 inches as the height, center of bearing to base of bracket.

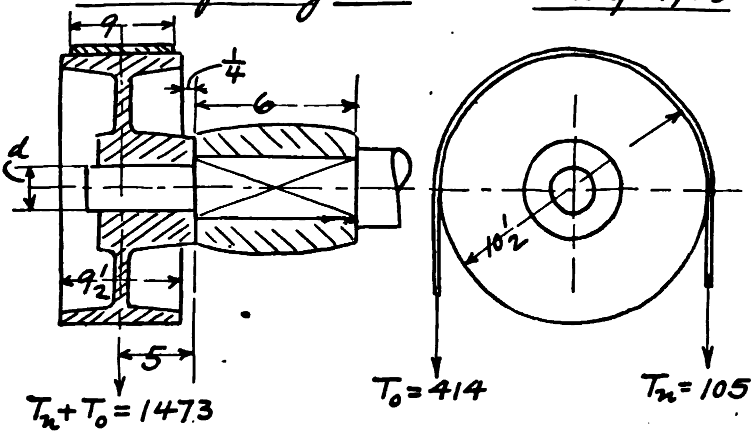
Data on Sketch. The data as found above should now be put on the sketch previously made; it will then have the appearance shown in Fig. 1.

This sketch is now in form to control all the subsequent detail design, and it is expected that the figured dimensions as shown can be maintained. It is impossible to predict this with positiveness, however, as in the working out of the minor details certain changes may be found desirable, when, of course, they should be made.

The shaft sizes do not appear on this sketch; hence before proceeding further the several shaft diameters must be calculated.

Sizes of Shafts. The calculations of the shaft diameters are good instances of systematic engineering computations, hence they are reproduced in the exact form in which they were made. The student should learn a valuable lesson in making and recording calculations by following these carefully. Note that each set of figures is independent, both in the statement of given data, as well as in the actual computation. Observe how easy it would be for the author of these figures or anyone else to check them even after

Small pulley bore. May 21, 03



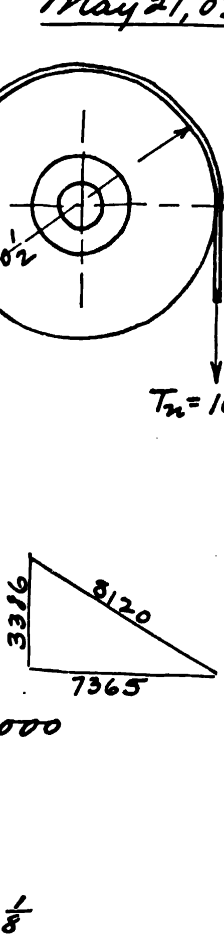
$B = 1473 \times 5 = 7365$

$T = 645 \times 5.25 = 3386$

$T_e = B + \sqrt{B^2 + T^2}$

$= 7365 + \sqrt{7365^2 + 3386^2}$

$= 7365 + 8120 = 15485$



$T_e = \frac{sI}{c} = \frac{sd^3}{5.1}$ Let $s = 8000$

$15485 = \frac{8000\,d^3}{5.1}$

$d^3 = \frac{15485 \times 5.1}{8000} = 9.872$

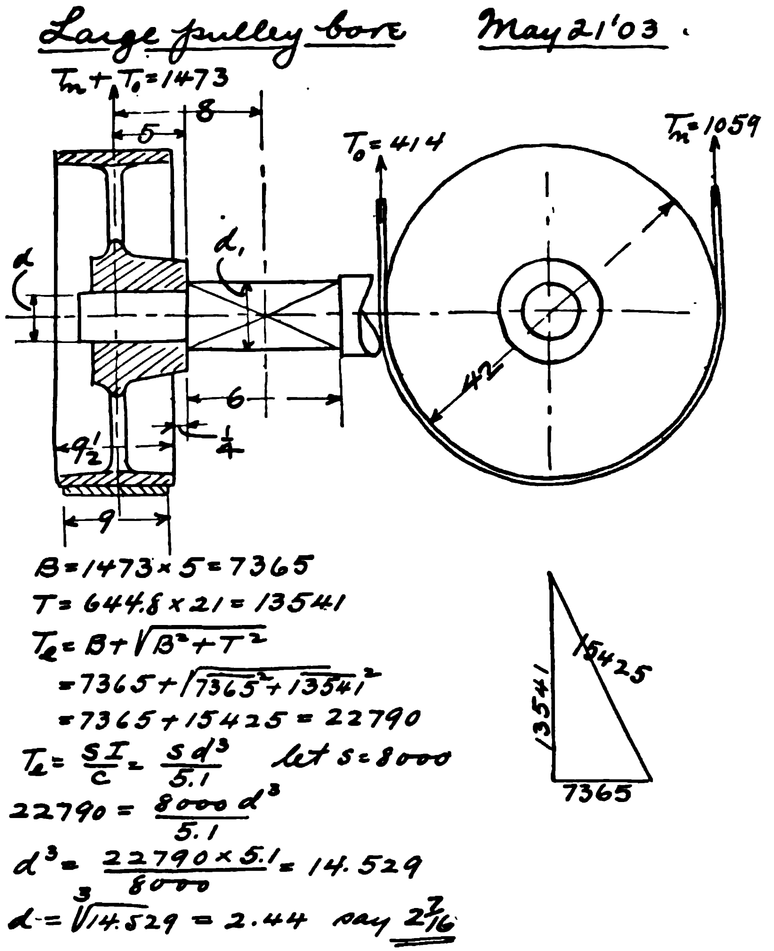
$d = \sqrt[3]{9.872} = 2.14$ say $2\frac{1}{8}$

Fig. 4.

a long lapse of time. If the machine should unexpectedly fail in service the figures are always available to prove or disprove theoretical weakness. The right triangles merely indicate that the value of $\sqrt{B^2+T^2}$ was found by the graphical method suggested in

Part II, "Shafts," the figures being put on the triangle as a simple and direct way of recording both process and result.

Attention is especially called to the fact that in the pinion shaft the size is changed for each piece upon the shaft. This is


Large pulley bore May 21'03.

$T_m + T_o = 1473$

$$B = 1473 \times 5 = 7365$$

$$T = 644.8 \times 21 = 13541$$

$$T_e = B + \sqrt{B^2 + T^2}$$

$$= 7365 + \sqrt{7365^2 + 13541^2}$$

$$= 7365 + 15425 = 22790$$

$$T_e = \frac{SI}{c} = \frac{Sd^3}{5.1} \quad \text{let } S = 8000$$

$$22790 = \frac{8000\,d^3}{5.1}$$

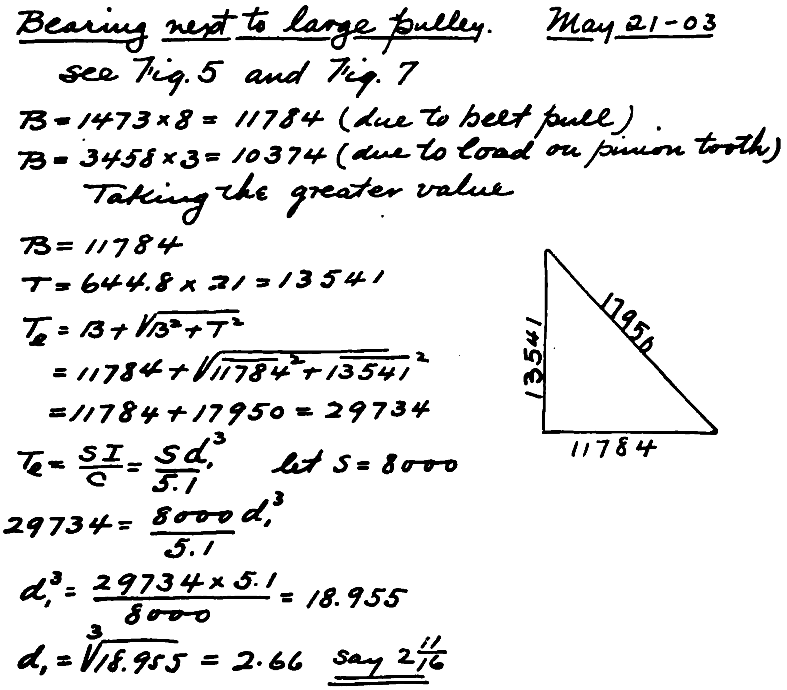
$$d^3 = \frac{22790 \times 5.1}{8000} = 14.529$$

$$d = \sqrt[3]{14.529} = 2.44 \quad \text{say } 2\tfrac{7}{16}$$

Fig. 5.

done partly because it is desired to show the student that the shaft at each of these points should be theoretically of different size. It is also done because as a practical feature of construction it is a good plan to change the size when the fit changes, partly for reasons of production in the shop, partly for ease in slipping pieces freely endwise on the shaft until they reach their proper fit and location in the assembling of the machine.

This should not be taken as an absolute requirement in any sense. A straight shaft would be satisfactory in the present case; but the shouldered shaft is a little better construction, in a mechanical sense, and does not cost much more. Hence it is used. For the drum the straight shaft seems to answer the requirement well enough.



Bearing next to large pulley. May 21-03

see Fig. 5 and Fig. 7

$B = 1473 \times 8 = 11784$ (due to belt pull).

$B = 3458 \times 3 = 10374$ (due to load on pinion tooth)

Taking the greater value

$B = 11784$

$T = 644.8 \times 21 = 13541$

$T_e = B + \sqrt{B^2 + T^2}$

$= 11784 + \sqrt{11784^2 + 13541^2}$

$= 11784 + 17950 = 29734$

$T_e = \frac{SI}{c} = \frac{Sd_1^3}{5.1}$ let $S = 8000$

$29734 = \frac{8000\, d_1^3}{5.1}$

$d_1^3 = \frac{29734 \times 5.1}{8000} = 18.955$

$d_1 = \sqrt[3]{18.955} = 2.66$ say $2\frac{11}{16}$

Fig. 6.

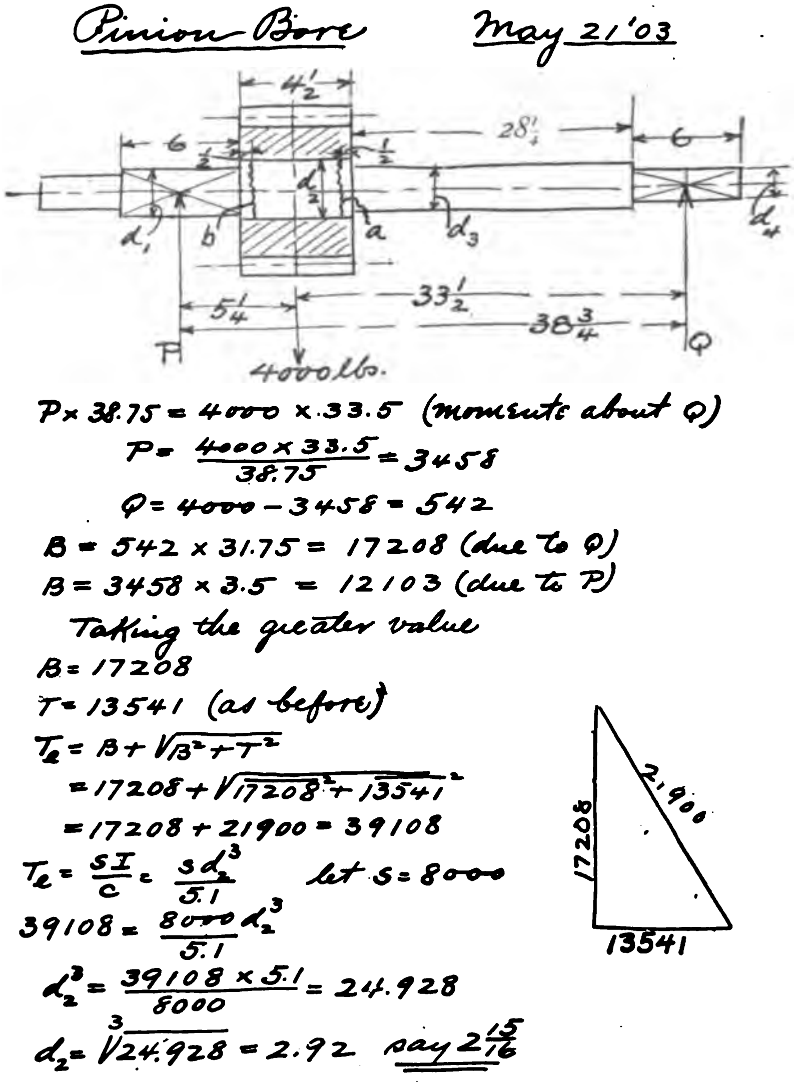
Small Pulley Bore. Fig 4.

Large Pulley Bore. Fig. 5.

Bearing Next to Large Pulley. Fig. 6.

The diameter, $2\frac{11}{16}$, as calculated, is based on the supposition that the greatest bending moment is caused by the belt pull on the overhanging pulley, that is, by the forces existing at the left-hand side of the center of the bearing.

But the pinion tooth load produces a heavy bending on the shaft in the bearing, the shaft in this case acting as a beam sup-



$4\frac{1}{2}$
$28\frac{1}{4}$
6
6
$\frac{1}{2}$
$\frac{1}{2}$
d_2
d_1
b
a
d_3
d_4
$5\frac{1}{4}$
$33\frac{1}{2}$
$38\frac{3}{4}$
P
Q
4000 lbs.
17208
21900
13541

Pinion Bore May 21 '03

$$P \times 38.75 = 4000 \times 33.5 \quad \text{(moments about } Q\text{)}$$

$$P = \frac{4000 \times 33.5}{38.75} = 3458$$

$$Q = 4000 - 3458 = 542$$

$$B = 542 \times 31.75 = 17208 \quad \text{(due to } Q\text{)}$$

$$B = 3458 \times 3.5 = 12103 \quad \text{(due to } P\text{)}$$

Taking the greater value

$$B = 17208$$

$$T = 13541 \quad \text{(as before)}$$

$$T_e = B + \sqrt{B^2 + T^2}$$

$$= 17208 + \sqrt{17208^2 + 13541^2}$$

$$= 17208 + 21900 = 39108$$

$$T_e = \frac{SI}{c} = \frac{S d_2^3}{5.1} \quad \text{let } S = 8000$$

$$39108 = \frac{8000 d_2^3}{5.1}$$

$$d_2^3 = \frac{39108 \times 5.1}{8000} = 24.928$$

$$d_2 = \sqrt[3]{24.928} = 2.92 \quad \text{say } 2\tfrac{15}{16}$$

Fig. 7.

ported at the two bearings and having the tooth load applied as shown. If this latter effect be greater than the former, that is, if the bending moment produced by the pinion tooth load be greater than the bending moment produced by the belt pull, then the diameter must be increased to satisfy the latter case. As is seen by the second calculation of Fig. 6, this is not the case, and the diameter stands at $2\frac{1}{16}$ as made.

Pinion Bore. Fig. 7. The pinion being a driving fit upon the shaft, reinforces the shaft to such an extent that it is hardly possible for the shaft to break off very far inside the face of the pinion; but it is quite possible that the metal of the pinion may give enough, or be a little free at the ends of the hole, so that the shaft may be broken off, say $\frac{1}{2}$ inch inside the face. In this case, it may fail from the moment of the force at the left-hand bearing or of that at the right. It may fail then at (a) or (b), depending on which section has the greater bending moment. Trying both, it is seen by the calculation that the right-hand moment is the controlling one, and it, therefore, is used.

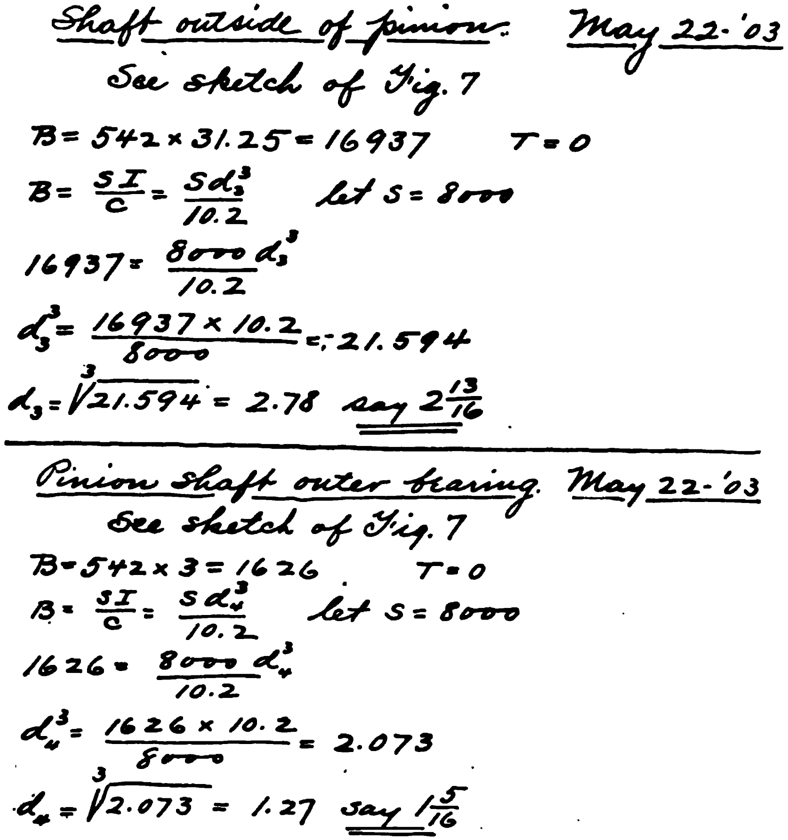
Shaft Outside of Pinion. Fig. 8. As there is no power transmitted through this portion of the shaft, there is no torsional moment in it, and the bending moment remains practically the same as inside the pinion.

The size figures about $2\frac{1}{16}$, but since there is no use in turning off material just to reduce the size to this, it is well to make it $2\frac{7}{8}$, or just smaller than the fit in the pinion.

Pinion Shaft Outer Bearing. Fig. 8. This diameter, of course, figures small, as there is no torsion in it, and the bending moment is not heavy. The practical question comes in, however, whether it is advisable to make the outer bracket different from the inner one just on account of this bearing. The commercial answer to this would probably be "No," hence the size as figured next to the pinion will be maintained ($2\frac{1}{16}$).

Drum Shaft. Fig. 9. In this case, as previously inferred, the simplest thing to do is to use a piece of straight cold-rolled steel, and make both bearings alike, the size being determined according to the worst case of loading which can occur as the rope travels from end to end of the drum. This case is evidently when the rope is at the end of its travel close to the brake, for at that time both the load on the rope and the load on the pinion tooth which is driving it are exerted upward, and produce the greatest reaction at the bearing next to the gear. The analysis of the forces for this condition is shown in Fig. 9.

Other conditions of loading would be when the brake is on and the tooth load relieved, but then the resultant of the brake strap tensions would be diagonally downward and would reduce



Shaft outside of pinion. May 22-'03

See sketch of Fig. 7

$B = 542 \times 31.25 = 16937$ $T = 0$

$B = \frac{SI}{c} = \frac{Sd_3^3}{10.2}$ let $S = 8000$

$16937 = \frac{8000\,d_3^3}{10.2}$

$d_3^3 = \frac{16937 \times 10.2}{8000} = 21.594$

$d_3 = \sqrt[3]{21.594} = 2.78$ say $2\frac{13}{16}$

Pinion shaft outer bearing. May 22-'03

See sketch of Fig. 7

$B = 542 \times 3 = 1626$ $T = 0$

$B = \frac{SI}{c} = \frac{Sd_4^3}{10.2}$ let $S = 8000$

$1626 = \frac{8000\,d_4^3}{10.2}$

$d_4^3 = \frac{1626 \times 10.2}{8000} = 2.073$

$d_4 = \sqrt[3]{2.073} = 1.27$ say $1\frac{5}{16}$

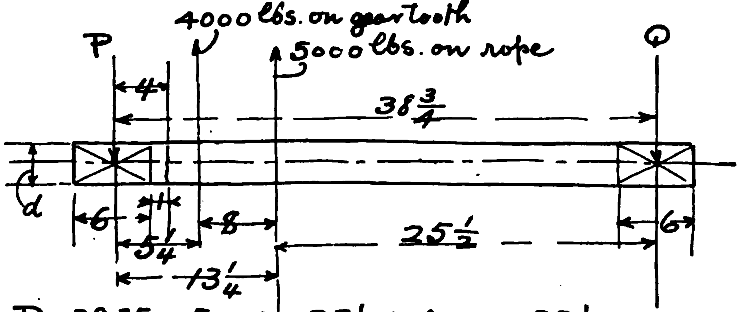
Fig. 8.

rather than add to the rope load. Again, when the rope is at the end of the drum farthest from the gear, the load on it and the load on the pinion tooth are both exerted upward as before, but the reaction cannot be as great as in the case of Fig. 9, because the tooth load is still concentrated at the other end of the shaft and produces a relatively small reaction at the rope end

Preliminary Layout. Fig. 10. Proceeding now with the layout to scale, the detail of the parts may be worked out as completely as the scale of the drawing will permit. The work on this drawing may be of an unfinished, sketchy nature, but the measurements must be exact as far as they go, for this drawing is to serve as the reference sheet, from which all future detail is to be worked up.

In this layout may be worked out the sizes of the arms and hubs of pulleys and gears, the proportions of the drum and brake

Drum shaft May 22'03



$$P \times 38.75 = 5000 \times 25\tfrac{1}{2} + 4000 \times 33\tfrac{1}{2}$$

$$P = \frac{261500}{38.75} = 6748$$

$$B = 6748 \times 4 = 26992$$

$$B = \frac{SI}{c} = \frac{Sd^3}{10.2} \quad \text{let } S = 8000$$

$$26992 = \frac{8000\,d^3}{10.2}$$

$$d^3 = \frac{26992 \times 10.2}{8000} = 34.415$$

$$d = \sqrt[3]{34.415} = 3.25 \quad \text{say } 3\tfrac{1}{4}$$

Fig. 9.

strap, and the general dimensions of the side brackets and the base. When the detail becomes too fine to work out to advantage on this drawing it may be worked out full size by a separate sketch, or left to be finished when it is regularly detailed. The preliminary layout, it should be remembered, is a service sheet only, a means of carrying along the design, and not intended for

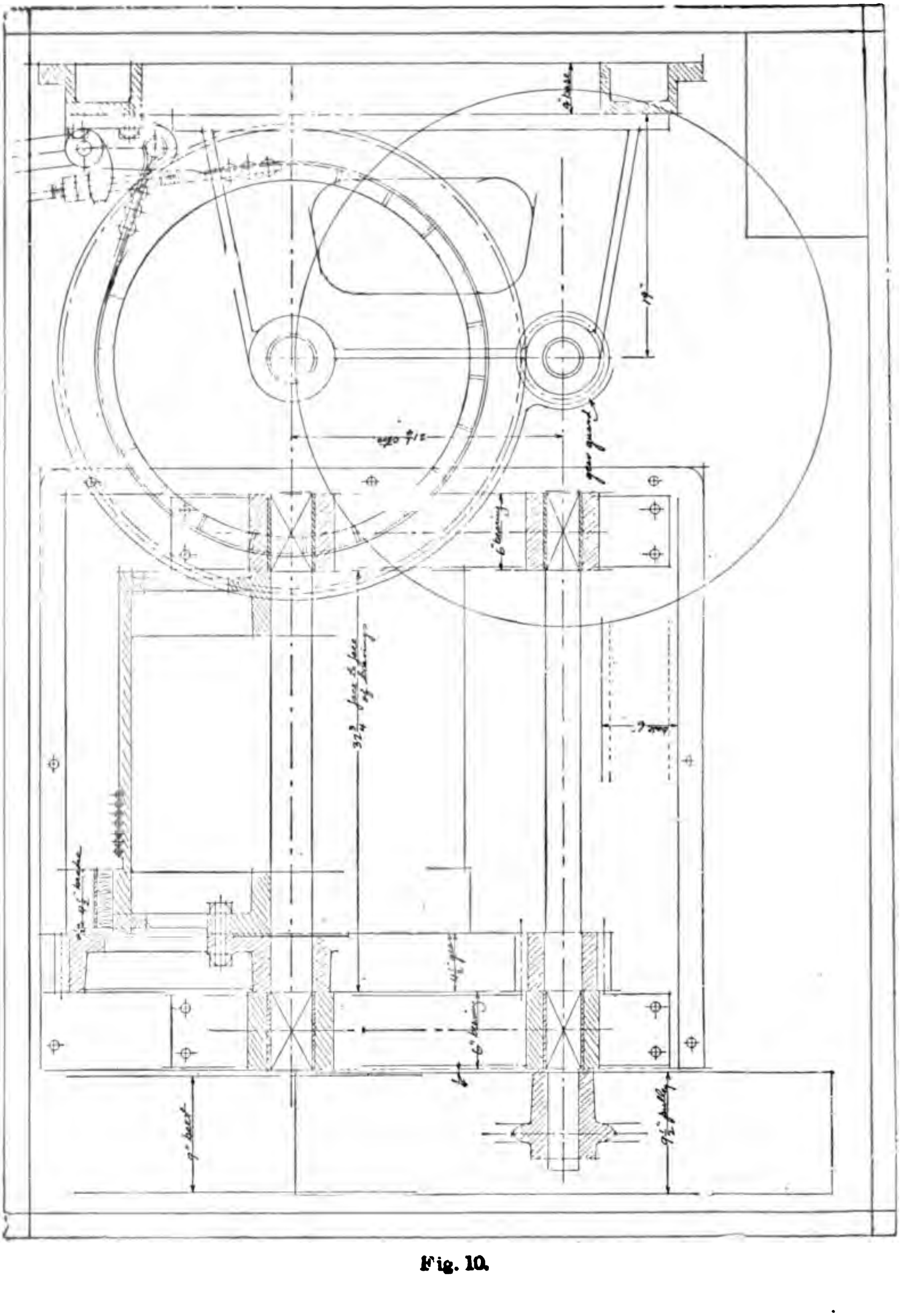

19"
21½" ctrs
gear guard
6" bearing
32¾" face to face of bearings
4½" gear
6" bearing
9" belt
9½" pulley

Fig. 10.

a finished drawing. The moment that the free use of the layout is impaired by trying to make too much of a drawing of it, its value is largely lost. A designer must have some place to try out his schemes and devices, and the layout drawing is the place to do it. This drawing may be recurred to at intervals in the progress of the design, details being filled in as they are worked out, as they may control the design of adjacent parts.

As the discussion of the design of each of the members involved in the present problem can be better taken up in connection with the detail drawing of each, it will be given there, rather than in connection with the layout, although many of the proportions thus discussed could be worked out directly from the latter.

Pulleys. Fig. 11. The analysis of the forces in the belt gives, according to the calculation of Fig. 3, a tension in the tight side of 1,059 pounds, and in the slack side 414 pounds. The difference of these, or 1,059—414=645 pounds, is transmitted to the pulley and produces the torque in the shaft. Of course in the small pulley the torque is transmitted from the motor through the pulley to the belt, but both cases are the same as far as the loading of the pulleys is concerned.

The only other force theoretically acting is the centrifugal force due to the speed of the pulley. This produces tension in the rim and arms, but for the low value of 1,300 feet per minute peripheral velocity in this case may be disregarded.

Considering the arms as beams loaded at the ends, and that one-half the whole number of arms take the load, and for convenience, figuring the size of the arms at the center of the pulley gives the following calculation for the large pulley:

$$\frac{645}{3}\times 21=\frac{S\times I}{c}=.0393\times 2{,}500\times h^3$$

Let $S=2{,}500$

" h=breadth of oval

" $.4h$=thickness of oval

$$h^3=\frac{4{,}515}{98.25}=46$$

$$h=\sqrt[3]{46}=3.6 \text{ (say 3.5)}$$

$$.4h=.4\times 3\ 5=1.4 \text{ (say 1 7-16)}$$

This is about all the theoretical figuring necessary on this pulley. The rim is made as thin as experience judges it capable of being cast; the arms are tapered to suit the eye, thus giving ample fastening to the rim to provide against shearing off the rim

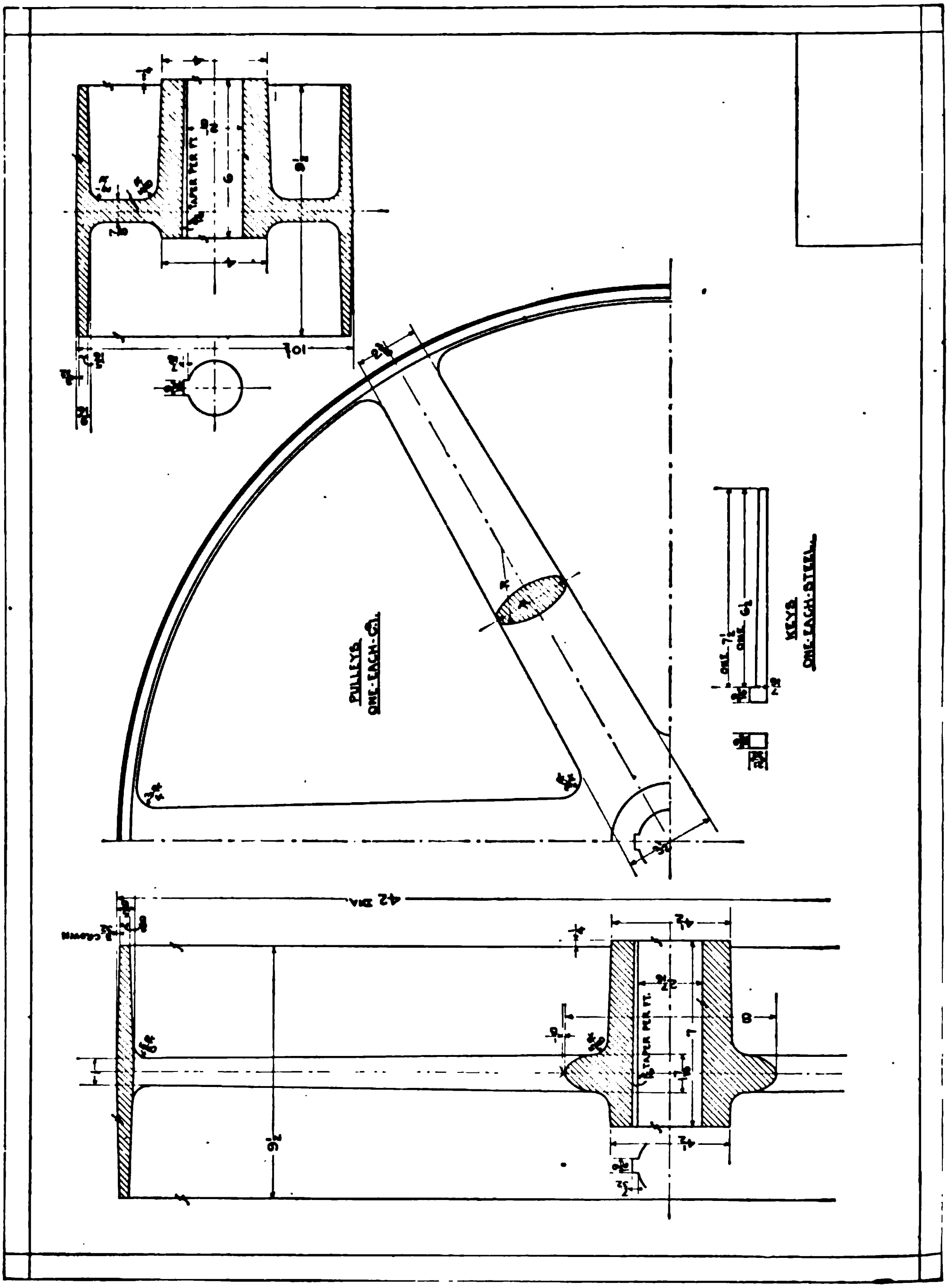

PULLEYS
ONE-EACH-C.I.
KEYS
ONE-EACH-STEEL
42 DIA
TAPER PER FT.
CROWN

Fig. 11

from the arms; generous fillets join the arms to both rim and hub; and the hub is given thickness to carry the key, and length enough to prevent tendency to rock on the shaft. Uncertain strains due to unequal cooling in the foundry mold may be set up in the arms and rim, but with careful pouring of the metal they should not be serious, and the low value chosen for the fibre stress allows considerable margin for strength.

The small pulley has the same forces to withstand as the large pulley, but on account of its small diameter there is not room enough for arms between the rim and the hub, hence it is made with a web. The web cannot be given any bending by the belt pull, the only tendency which exists in this case being a shearing where the web joins the hub. This shearing also exists throughout the web as well, but at other points farther from the center it is of less magnitude, and moreover, there is more area of metal to take it. The natural way to proportion the thickness of the web is to give it an intermediate thickness between that of the hub and rim, thus securing uniform cooling, and then figure the stress as a check. Making this value $\frac{7}{8}$ inch gives a shearing area of $\frac{7}{8}$ multiplied by the circumference of the hub, which is 3.1416 $\times 4 = 12.56$. The shearing force at the hub is $\frac{645\times 5.25}{2} = 1{,}693$ pounds. Equating the external force to the internal resistance

$$1{,}693 = \tfrac{7}{8}\times 12.56\times S$$

$$S = \frac{1{,}693\times 8}{7\times 12.56} = 154 \text{ pounds per square inch (approx.).}$$

This is a very low figure, even for cast iron, hence the web is amply strong. The rim and hub are proportioned as for the large pulley.

The keys are taken from the standard list. They may be checked for shear, crushing in the hub, and crushing in the shaft, but the hubs are so long that it is at once evident without figuring that the stress would run very low in both cases.

Gears. Fig. 12. The analysis of the forces acting on the gears has been given on page 28, 4,000 pounds being taken at the pitch line. Using this same value, and choosing a T-shaped arm as a good form for a heavily loaded gear like the present one, let us consider that the rim is stiff enough to distribute the load

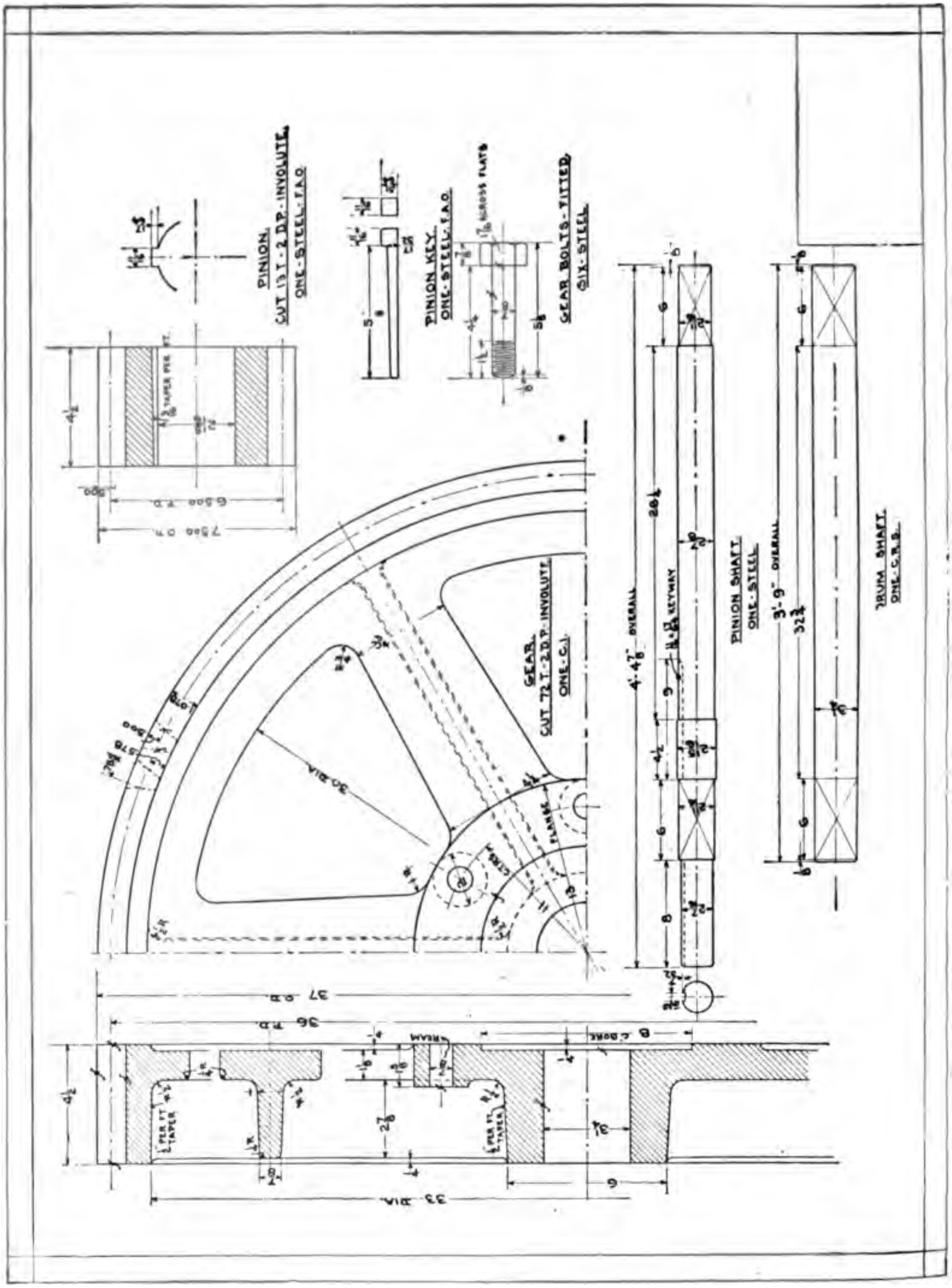

PINION.
CUT 13 T - 2 D.P. - INVOLUTE.
ONE - STEEL - F.A.O.
7.500 O.D.
6.500 P.D.
TAPER PER FT.
PINION KEY.
ONE - STEEL - F.A.O.
ACROSS FLATS
GEAR BOLTS - FITTED.
SIX - STEEL.
GEAR.
CUT 72 T - 2 D.P. - INVOLUTE.
ONE - C.I.
30 DIA.
FLANGE
CTRS
37 O.D.
36 P.D.
33 DIA.
8 BORE
REAM
PER FT. TAPER
4'-4 7/8" OVERALL
28 1/4
KEYWAY
PINION SHAFT.
ONE - STEEL
3'-9" OVERALL
32 3/4
DRUM SHAFT.
ONE - C.R.S.

Fig. 12.

equally between all the arms, and that each acts as a beam loaded at the end with its proportion of the tooth load. Before we can determine the length of these arms, however, we must fix upon the size of the flange which is to carry the driving bolts. This is taken at 13 inches. It could be smaller if desired, but drawing the bolts in toward the center increases the load on them, and 13 inches seems reasonable until it is proved otherwise. This makes the maximum moment which can come on an arm $\frac{4{,}000 \times 11.5}{6} = 7{,}666$ inch-pounds.

Now it is evident that the base of the T arm section, which lies in the plane of rotation, is most effective for driving, and that the center leg of the T does not add much to the driving capacity of the arm, although it increases the lateral stiffness of the arm, as well as providing in casting a free flow of metal between the rim and the hub. Hence the simplest way of treating the section of the arm for strength is to consider the base of the T only, of rectangular section, breadth b, and depth h, for which the internal moment of resistance is $\frac{S \times b \times h^2}{6}$.

Also, it is simplest to assume one dimension, say the breadth, and the allowable fibre stress, and figure for the depth. Taking the breadth at $1\frac{1}{8}$ inches, which looks about right, and the fibre stress at 2,500, and equating the external moment to the internal, we have

$$7{,}666 = \frac{2{,}500 \times 1.125 \times h^2}{6}$$

$$h^2 = \frac{6 \times 7{,}666}{2{,}500 \times 1.125} = 16.4$$

$$h = \sqrt{16.4} = 4.05 \text{ (say } 4\tfrac{1}{4})$$

Drawing in this size, and tapering the arm to the rim as in the case of the pulleys, making the depth of the rim according to the suggested proportions given in Part II, "Gears," giving the center leg of the T a thickness of $\frac{7}{8}$ inch tapering to 1 inch, and heavily filleting the arms to the rim and center flange, we have a fairly well proportioned gear.

The next thing to determine is the size of the driving bolts. The circle upon which their centers lie may be 11 inches in diam-

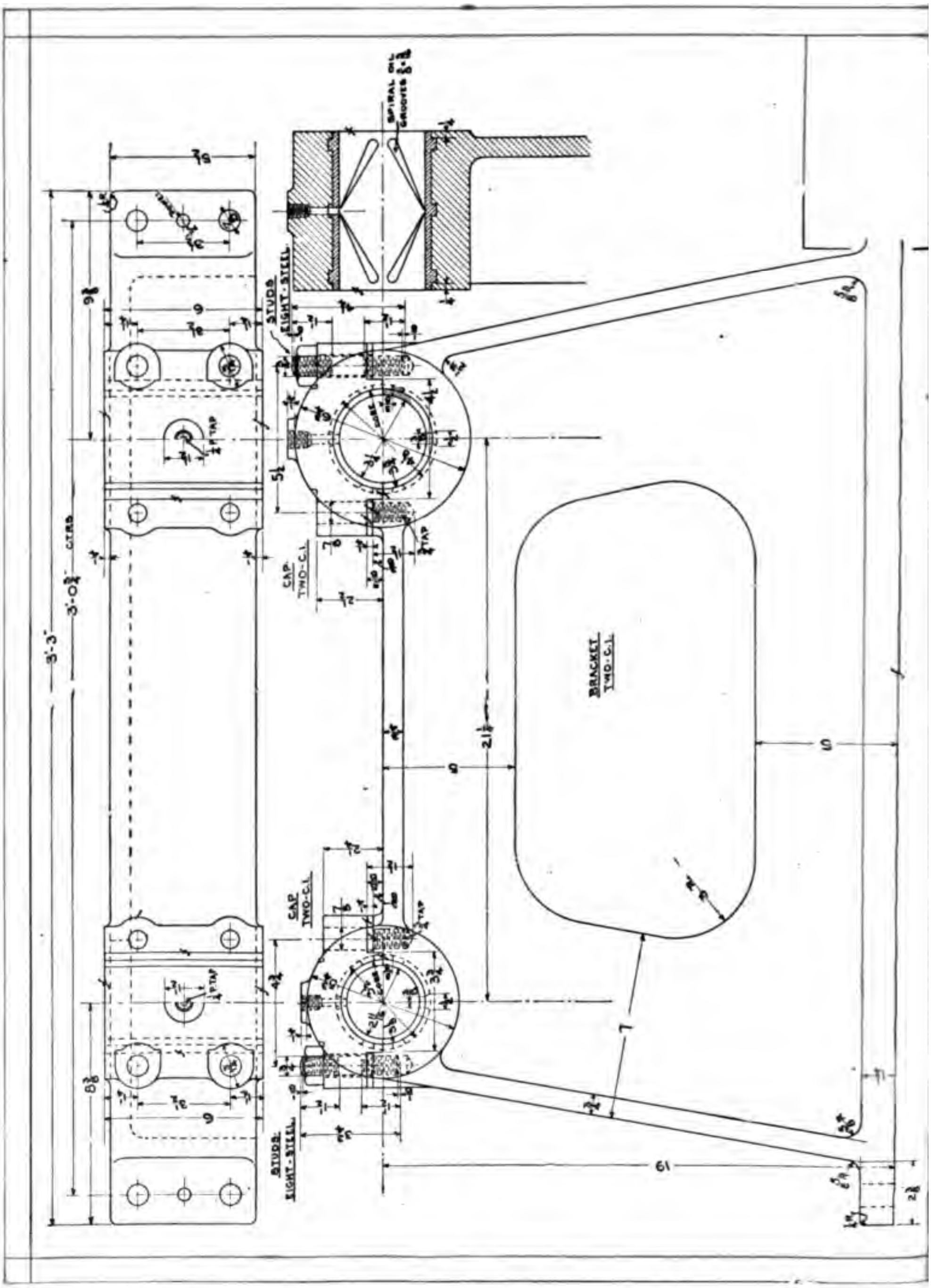

3'-3"
3'-0¼" CTRS.
STUDS EIGHT-STEEL
CAP TWO-C.I.
BRACKET TWO-C.I.
SPIRAL OIL GROOVES
21½
19
10

Fig. 13

eter, and there will naturally be six bolts, one between each arm. These bolts are in pure shear, and the material of which they are to be made ought to be good for at least 8,000 pounds per square inch fibre stress. The force acting at the circumference of an 11-inch circle would be $\frac{4{,}000\times18}{5.5}=13{,}091$ pounds.

Equating the load on each bolt to the resisting shear gives

$$\frac{13{,}091}{6}=8{,}000\times A=\frac{8{,}000\times3.1416\times d^2}{4}$$

Let A=area resisting shear.
Let d=dia. of bolt.
Then $A=\frac{\pi\, d^2}{4}$

$$d^2=\frac{4\times13{,}091}{6\times8{,}000\times3.1416}=.35$$

$d=\sqrt{.35}$ (say .6) $\frac{5}{8}$-inch bolts would do.

But $\frac{5}{8}$-inch bolts are pretty small to use in connection with such heavy machinery. They look out of proportion to the adjacent parts. Hence $\frac{7}{8}$-inch bolts have been substituted as being better suited to the place in spite of the fact that theoretically they are larger than necessary. The extra cost is a small matter. These bolts may crush in the flange as well as shear off, but as there is an area of $\frac{7}{8}\times1\frac{5}{8}=1.422$ square inches to take $\frac{13{,}091}{6}=2{,}182$ pounds, the pressure per square inch of projected area is only $\frac{2{,}182}{1.422}=1{,}534$ pounds, which is very low.

This gear needs no key to the shaft because all the power comes down the arms and passes off to the drum through the bolts, thus putting no torsional stress in the shaft. The face of the flange is counterbored so as to center the gear upon the drum, without relying upon the fit of the gear upon the shaft to do this.

The pinion is solid and needs no discussion for its design.

Brackets and Caps. Fig. 13. As the size of the drum shaft was determined by considering the rope wound close up to the brake, thus giving in combination with the load on the gear tooth the maximum reaction at the bearing as 6,748 pounds, the cap and bolts should be designed to carry the same load.

For a bearing but 6 inches long, two bolts are sufficient under ordinary conditions and might perhaps do for this case. The load is pretty heavy, however, and it is deemed wise to provide four bolts, thus securing extra rigidity, and permitting the use of bolts of comparatively small size. If the load were distributed equally over all the bolts each would take one-fourth of the whole load, but it is not usually safe to figure them on this basis, because it is difficult to guarantee that each bolt will receive its exact share of stress. Assuming that the two bolts on one side take ⅔ the whole load instead of ½, which provides for this uncertain extra stress, each bolt must take care of ⅓ of 6,748, or 2,249, pounds. Allowing 8,000 pounds per square inch fibre stress calls for an area at the root of the thread of $\frac{2,249}{8,000} = .281$ square inch. Consulting a table of bolts we find that the next standard size of bolt greater than this is ¾, which gives an area of .302 square inch.

Choosing this size as satisfactory, the bolts should be located as close to the shaft as will permit the hole to be drilled and tapped without breaking out. A center distance of 5½ inches accomplishes this result. The distance between centers in the other direction is somewhat arbitrary, although the theoretical distance between the bolt and the end of the bearing to give equal bending moment at the center of the cap and at the line of the bolts is about $\frac{5}{24}$ of the length, or $\frac{5}{24}$ of $6 = 1\frac{1}{4}$ inches. This proportion answers well for the present case, although for long caps it brings the bolts too far in to look well.

The thickness of the cap may be determined by assuming it to be a beam supported at the bolts and loaded at the middle. This is not strictly true, for the load is distributed over at least a portion of the shaft diameter; moreover, the bolts to some extent make the beam fixed at the ends. It being impossible to determine the exact nature of the loading, we may take it as stated, supported at the ends and loaded in the middle, and allow a higher fibre stress than usual, say 3,500. The longitudinal section at the middle of the cap is rectangular, of breadth 6 inches, and depth unknown, say h. The equation of moments is

$$\frac{W \times l}{4} = \frac{S \times I}{c} = \frac{S \times b \times h^2}{6}$$

$$\frac{6,748 \times 5.5}{4} = \frac{3,500 \times 6 \times h^2}{6}$$

$$h^2 = \frac{6 \times 6,748 \times 5.5}{4 \times 3,500 \times 6} = 2.65$$

$$h = \sqrt{2.65} = 1.62 \text{ (}1\tfrac{1}{2}\text{ will probably answer)}$$

For the other bearing next to the pinion, the load on the tooth acts downward, and the resultant pull of the belt is nearly horizontal, hence the cap and bolts must stand but little load, and calculation would give minute values. In a case like this it is well to make the size the same as for the larger bearing, unless the construction becomes very clumsy thereby. This saves changing drills and taps in making the holes, and preserves the symmetry of the bracket. The ¾-inch bolts are good proportion for the smaller bearing, hence that size will be maintained throughout.

The body of the bracket is conveniently made with the web at the side and horizontal ribs extending to the outside. The load due to the rope is carried directly down the side ribs and web into the bottom flanges and to the bolts. The analysis of the forces on these bolts is shown in Fig. 14. It is evident from the figure that the resultant belt pull tends to hold the bracket down, while the load on the rope tends to pull it up, the point about which it tends to rotate being the corner furthest from the drum. It is also evident that the bolts nearest this corner can have little effect on the holding down, because their leverage is so small about the corner, hence we shall assume that the pair of bolts at the right-hand end of the bracket takes all the load. The belt pull, being horizontal, tends to slide the bracket along the base, but this tendency is small, and at any rate is easily taken care of by the two dowel pins, which are thus put in shear.

The load on the bolts being 4,954 pounds, a heavy bending moment is thrown on the flange of the bracket, tending to break it off at the root of the fillet. The distance to the root of the fillet is ¾ inch; the section tending to break is rectangular, of breadth 5½ inches, and unknown depth h. The equation of moments is

$$W \times l = \frac{S \times I}{c} = \frac{S \times b \times h^2}{6}$$

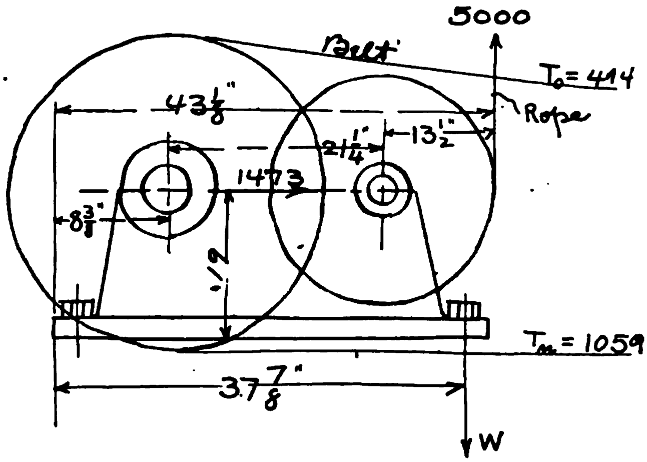
$$\frac{4{,}954 \times 3}{4} = \frac{2{,}500 \times 5.5 \times h^2}{6}$$

$$h^2 = \frac{6 \times 4{,}954 \times 3}{4 \times 2{,}500 \times 5.5} = 1.62$$

$$h = \sqrt{1.62} = 1.3 \text{ (say } 1\tfrac{1}{4}\text{).}$$

The thickness of the web and ribs of this bracket is hardly capable of calculation. The figure ¾ inch has been chosen in proportion to the size of the large drum bearing, giving ample stiffness and rigidity, and permitting uniform flow and cooling of the metal in the mold. The opening in the center is made merely to save material, as in that part little stress would exist, the two sides

Bracket base bolts May 25-1903



$$5000 \times 43\tfrac{1}{8} = 37\tfrac{7}{8} \times W + 1473 \times 19$$

$$W = \frac{5000 \times 43.125 - 1473 \times 19}{37.875}$$

W = 4954 lbs on two bolts

$\frac{4954}{2} = 2477$ lbs per bolt.

$\frac{2477}{8000} = .31$ sq. in. area at 8000 lb. stress.

Area at root of thread for a ¾ bolt is .28

Hence ¾ bolts will answer.

Fig. 14.

carrying the load down to the base bolts, and the top serving as a tie between the bearings.

This bracket might be made with the web in the center of the bearings instead of at the side, in which case the expense of the pattern would be slightly greater. It could also be made of closed box form, but would in that case probably weigh more than as shown.

Drum and Brake. Fig. 15. The analysis of the forces acting on the drum is simple, but its theoretical design is more complicated. It is evident that the drum acts as a beam of hollow circular cross section, and that its worst case of loading is when the rope is at or near the middle of the drum length. At the same time the metal of this circular cross section is in a state of torsion between the free end of the rope and the driving gear, due to the load on the gear tooth and the reaction of the rope. Also the wrapping of the rope around the drum tends to crush the metal of the section beneath it, the maximum effect of this action being near the free end of the rope where its tension has not been reduced by friction on the drum surface.

Now the "mechanics" to solve the problem of these three combined actions is rather complicated. It can be at least approximately solved, however, for it satisfies fairly well the case of combined compression and shear. But on a further study of this particular case, it is seen at once that the diameter of the drum is relatively large with respect to its length, which means that the thickness of the metal may be very small and yet give a large resisting area, or value of "I," both in direct bending as well as torsion; also it is so short that the external bending moment will be small. The practical condition now comes in, that the drum can be safely cast only when the thickness of the metal is at a minimum limit, for the core may be out of round, not set centrally, or by some other variation produce thin spots or even develop holes reaching out into the rope groove, discovered only when the latter is turned in the lathe.

Hence it seems reasonable and safe in this case to make the thickness of the drum depend simply upon the crushing caused by the wrapping of the rope around it, and we shall take the coil nearest the free end of the rope, assuming that it carries the full load of 5,000 pounds throughout one complete wrap around the drum.

The area resisting the crushing action may be considered to be that of the cross section of a ring, of width equal to the pitch

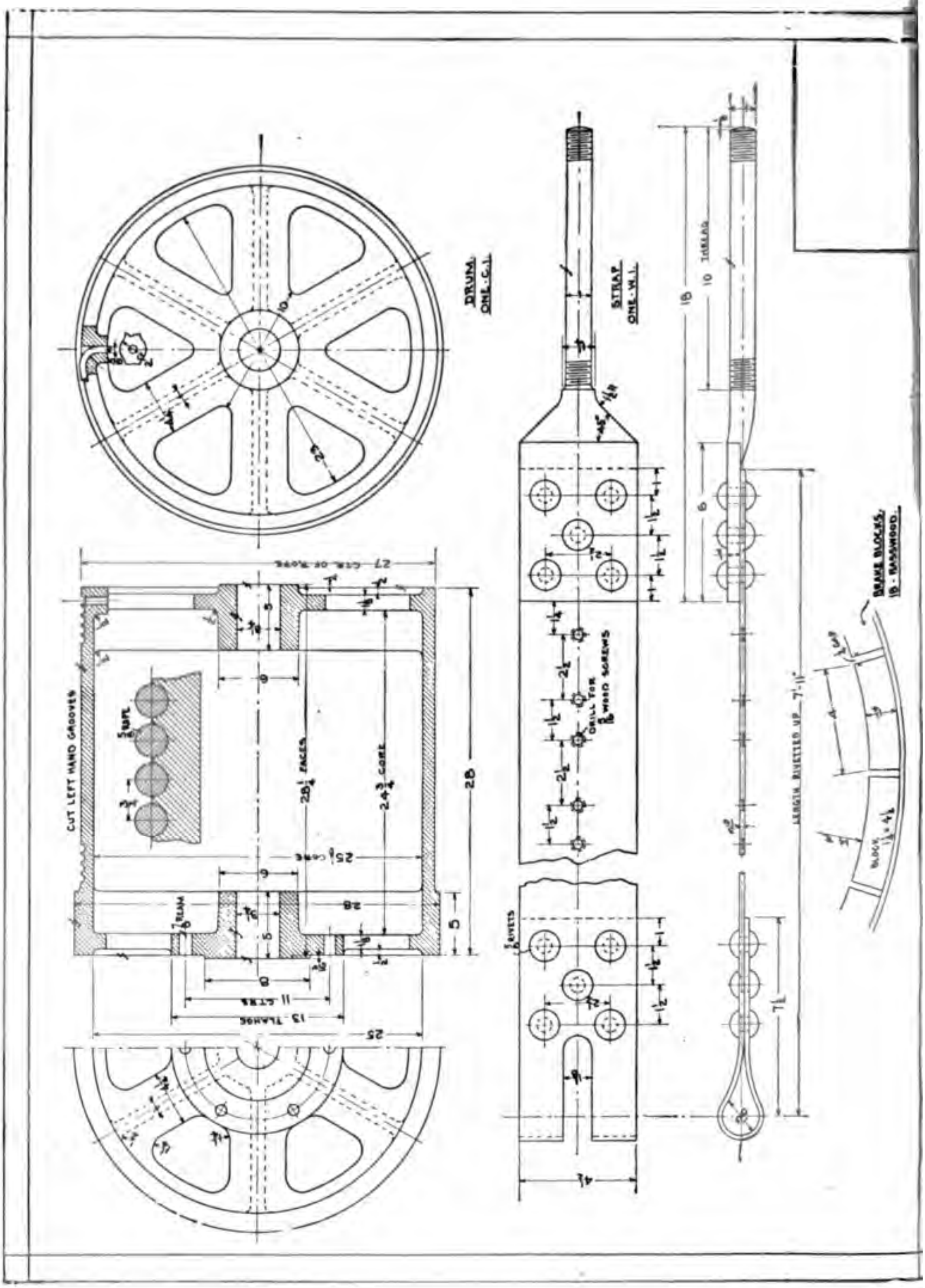

DRUM.
ONE-C.I.
STRAP
ONE-W.I.
BRAKE BLOCKS.
18-BASSWOOD.
CUT LEFT HAND GROOVES
27 CTR. OF ROPE
28¼ FACES
24¾ CORE
25¼ CORE
28
25
13 FLANGE
11 CTRS.
DRILL FOR ⅛ WOOD SCREWS
RIVETS
LENGTH RIVETTED UP 7'-11"
10 THREAD
18
6
7½
4½
BLOCK

Fig. 15.

of the groove. Assuming that $\frac{5}{8}$ inch is the least thickness which can be safely allowed under the groove for casting purposes, let us figure the crushing fibre stress to see if this is sufficiently strong. Disregarding the small amount of metal existing above the bottom of the groove, this gives the area to resist the crushing $\frac{5}{8} \times \frac{3}{4} = \frac{15}{32}$, or .47 inch. Since there are two of these sections and the rope acts on both sides, the equation of forces is:

$$5{,}000 \times 2 = S \times .47 \times 2$$

$$S = \frac{5{,}000 \times 2}{.47 \times 2} = 10620 \text{ pounds per square inch.}$$

This, for cast iron, in pure crushing, allows plenty of margin for the extra bending and torsional stress, which for such a considerable thickness would be slight.

The above case indicates a method of reasoning much used in designing machinery, which while following out the specified routine of thought as previously given in these pages, stops short of elaborate and minute theoretical calculation when such is obviously unnecessary. If a drum of great length were to be designed, and of small diameter, the same method of reasoning would deduce the fact that the design should be based on the bending and the torsional moments, the thickness in such a case being so great to withstand these that the intensity of the crushing due to wrap of the rope becomes of inappreciable value.

The remaining points of design of the drum are determined from practical considerations and judgment of appearance. The ribs behind the arms are put in to give lateral stiffness and guard against endwise collapse. The arms are subject to the same bending as those of the gear, but as they are equally heavy it is not necessary to calculate them. The flange at the driving end is of course matched to that already designed for the gear. The rope is intended to be brought through the right-hand end with an easy bend and the standard form of button wedged on to prevent its pulling through.

This drum would probably be cast with its axis vertical, and the driving flange down to secure sound metal at that point. Heavy risers would be left at the other end to secure soundness where the rope is fastened. Drums are often cast with the axis horizontal, but the vertical method is more certain to produce a sound casting. The grooves should be turned from the solid metal, partly because it is a difficult matter to cast them, but principally because the rope should run on as smooth surface as possible to avoid undue wear. On drums which carry chain instead of wire rope the grooves are sometimes cast with success, although even in this case the turned groove is generally preferable.

The brake consists of a wrought-iron band to which are fastened wooden blocks, the iron band giving the requisite strength while the blocks give frictional grip on the drum surface and can be easily replaced when worn. As in the designing of a belt the object in view is the grip on the pulley surface by the leather to enable power to be transmitted from the belt to the pulley, so in the case of the brake if we put the proper tension in the strap it can be made to grip the brake drum so tightly that motion between it and the drum cannot occur. The latter case is really the reverse of the first, if the driven pulley be considered, but is identical with the case of the driving pulley, in which the power is transmitted from the pulley to the belt. Of course in the case of the brake no power is transmitted, as when the brake holds no motion occurs, but the principle of the relative tensions in the strap is the same as for the belt.

Since the brake drum surface is 28 inches in diameter, the load at that surface which the brake must hold is

$$P = \frac{5{,}000 \times 27}{14 \times 2} = 4{,}821 \text{ pounds.}$$

We have then the following calculation corresponding exactly to that of the belt given in Fig. 3.

$\log. \frac{T_n}{T_o} = 2.729 \times \mu \times n$ Let $\mu = .25$

" $n = .75$

$T_n - T_o = P = 4{,}821$

$\log. \frac{T_n}{T_o} = 2.729 \times .25 \times .75 = 0.512$ (for which the natural number is 3.25).

Then $\frac{T_n}{T_o} = 3.25$ $T_o = \frac{T_n}{3.25}$

$T_n - T_o = 4{,}821$ $T_n - \frac{T_n}{3.25} = \frac{2.25 \times T_n}{3.25} = 4{,}821$

$T_n = \frac{4{,}821 \times 3.25}{2.25} = 6{,}963$ pounds (say 7,000)

$T_o = 6{,}963 - 4{,}821 = 2{,}142$ pounds (say 2,200)

The tight end of the strap must then be capable of carrying a load of 7,000 pounds, and since the width has already been taken at $4\frac{1}{2}$ inches, the problem is to find the necessary thickness. Equating the external load to the internal resistance we have

$$7{,}000 = A \times S \qquad \text{Let } t = \text{thickness}$$

$$\text{“ } S = \text{fibre stress} = 12{,}000$$

$$7{,}000 = 4.5 \times t \times 12{,}000$$

$$t = \frac{7{,}000}{4.5 \times 12.000} = .13 \text{ inch}$$

This, however, can be but a preliminary figure, for the riveting of the strap will take out some of the effective area, and the thickness will have to be increased to allow for this. Suppose on the basis of this figure we assume the thickness at a slightly increased value, say $\frac{3}{16}$ inch, and proceed to calculate the rivets.

A group of five rivets will work in well for this case, which gives $\frac{7{,}000}{5} = 1{,}400$ pounds per rivet. A safe shearing fibre stress is 6,000, hence the area necessary per rivet is $\frac{1{,}400}{6{,}000} = .23$ square inch. This comes nearest to the area $\frac{9}{16}$ diameter, but for the sake of using the more general size of rivet ($\frac{5}{8}$ inch) the latter is chosen, for which the area is .30.

We must now try these rivets in a $\frac{3}{16}$-inch plate for their safe bearing value. The projected area of a $\frac{5}{8}$-inch hole in a $\frac{3}{16}$-inch plate is $\frac{5}{8} \times \frac{3}{16} = .117$ square inch. $\frac{1{,}400}{.117} = 11{,}965$ (15,000 would be safe)

Taking out two $\frac{5}{8}$-inch rivets from the full width of $4\frac{1}{2}$ inches leaves $4\frac{1}{2} - (2 \times \frac{5}{8}) = 3.25$, and makes the net area of strap to take stress $3.25 \times \frac{3}{16} = .61$ square inches. Re-calculating the fibre stress for this area gives

$$7{,}000 = .61 \times S$$

$$S = \frac{7{,}000}{.61} = 11{,}475 \text{ (which approximates the previous value of 12,000).}$$

The slack end of the strap has to take but 2,200 pounds, hence a different calculation might be made for this end giving smaller rivets; but as it is impractical to change the thickness of the strap to meet this reduced load, it is well to maintain the same proportion of joint as at the tight end. The spacing of the rivets in both

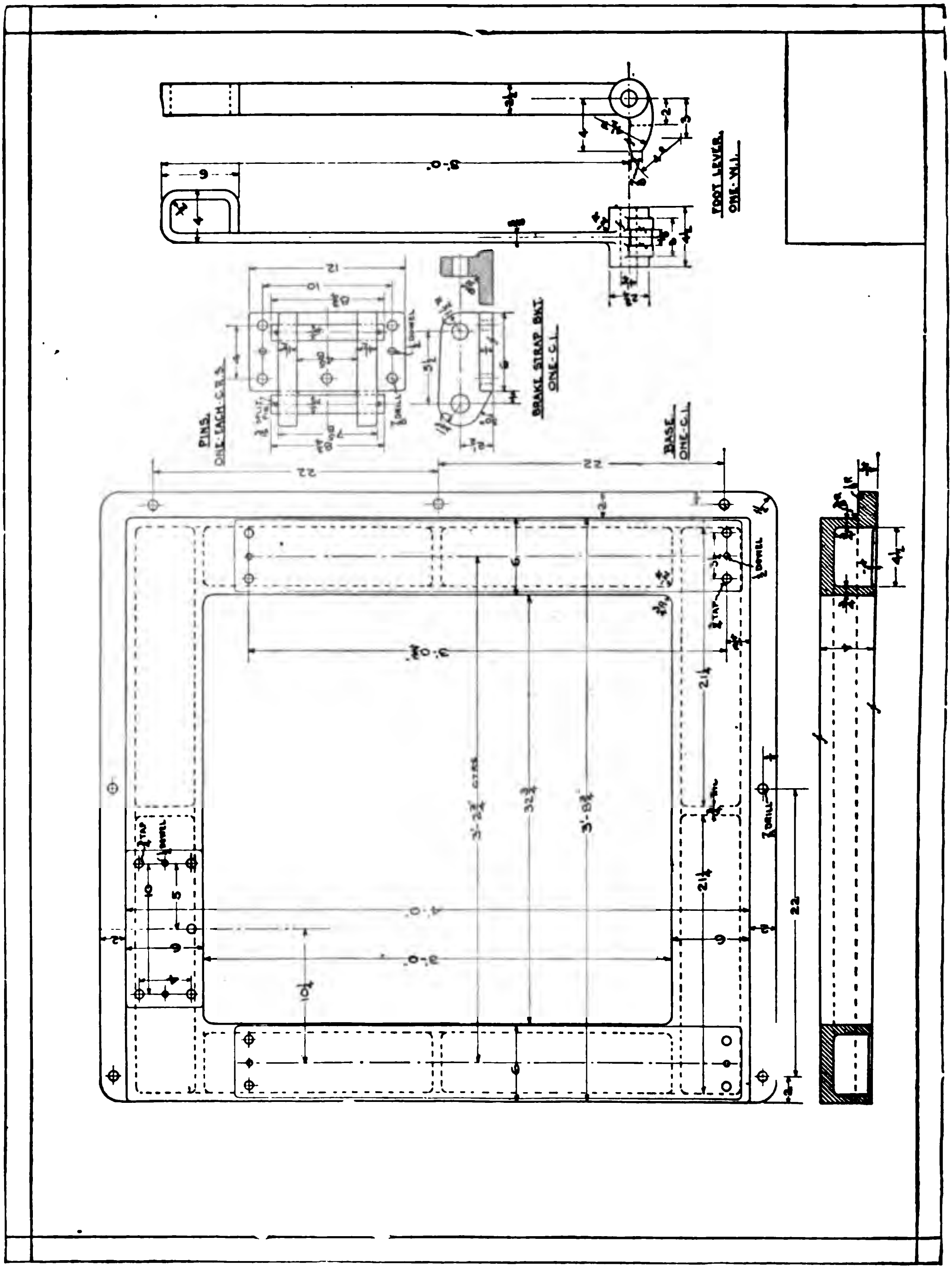

FOOT LEVER.
ONE-W.I.
BRAKE STRAP BKT.
ONE-C.I.
PINS.
ONE-EACH-C.R.S.
BASE.
ONE-C.I.
3'-0"
22
12
10
3'-2¼" CTRS
32¾
3'-8¾"
4'-0"
3'-0"
21¼
10¼
5
6
2
4½
⅞ DRILL
½ TAP
¼ DOWEL

Fig. 16.

cases follows the ordinary rule allowing at least three times the diameter of the rivet as center distance, and one-half this value to the edge of the plate.

The threaded end of the forging on the strap also has to carry the load of 2,200 pounds, for which a size smaller than 1 inch would suffice. It is natural, however, for the sake of general proportion to make the bolt as strong as the strap, and a 1-inch bolt gives an area of .52 square inch, nearly equalling the value of .61 net area of strap noted above.

Base, Brake-Strap Bracket and Foot Lever. Fig 16. The base cannot be definitely calculated, and can best be proportioned by judgment. It must not distort, twist, or spring in any way to throw the shafts out of line. The area in contact with the foundation upon which it rests must be ample to carry the weight of the whole machine with a low unit pressure. Although the form shown is perfectly practicable to cast and machine, and is simple and rigid, yet it is questionable if a bolted-up construction, say of four pieces, might not be equally rigid and yet involve greater facility of production in both the foundry and machine shop on account of the reduced sizes of parts to be handled. This is a question which depends on the equipment and methods of the individual shop, and is an illustration of the practical control of design by manufacturing conditions.

The brake-strap bracket and foot lever, also shown in this figure, are examples of machine parts which are quite definitely loaded, and the designing of which is a simple matter. Further discussion of their design is not made, the student being given opportunity for some original thought in determining the forces and moments that control their design.

Gear Guard and Brake-Relief Spring. In exposed machinery of this character it is desirable to cover over the gears with a guard to prevent anything accidentally dropping between the teeth and perhaps wrecking the whole machine. This guard is not shown, as it involves little of an engineering nature to interest the student. It could readily be made of sheet metal or light boiler plate, bent to follow the contour of the gears and fastened to the top flange of the main bracket.

If the brake be not automatically supported at its top it will lie with considerable pressure, due to its own weight, on the brake surface when it is supposed to be free from it, and by the friction thereby created will produce a heavy drag and waste of power. A spring connection fastened to an overhead beam is a simple way of accomplishing the desired result. A flat supporting strap carried out from the gear guard, having some degree of spring in it, is a neater method of solving the problem. The spring should be just strong enough to counterbalance the weight of the strap and yet not resist to an appreciable degree the force applied to throw the brake on.

GENERAL DRAWING.

The last step in the process of design of a machine is the making of the assembled or general drawing. This should be built up piece by piece from the detail drawings, thereby serving as a last check on the parts going together. This drawing may be a cross section or an outside view. In any case it is not wise to try to show too much of the inside construction by dotted lines, for if this be attempted, the drawing soon loses its character of clearness, and becomes practically useless. A general drawing should clearly **hint at, but not specify,** detailed design. It is just as valuable a part of the design as the detail drawing, but it cannot be made to answer for both with any degree of success. A good general drawing has plenty of views, and an abundance of cross sections, but few dotted lines.

The general drawing of the machine under consideration is left for the student to work up from the complete details shown. It would look something like the preliminary layout of Fig. 10, if the same were carefully carried out to finished form. A plain outside view would probably be more satisfactory in this case than a cross section, as the latter would show little more of value than the former. The functions which the general drawing may serve are many and varied. Its principal usefulness is, perhaps, in showing to the workman how the various parts go together, enabling him to sort out readily the finished detail parts and assemble them, finally producing the complete structure. Otherwise the making of a machine, even with the parts all at hand, would be like the putting together of the many parts of an intricate puzzle, and much time would be wasted in trying to make the several parts fit, with perhaps never complete success in giving each its absolutely correct location.

The general drawing also gives valuable information as to the total space occupied by the completed machine, enabling its location in a crowded manufacturing plant to be planned for, its connection to the main driving element arranged, and its convenience of operation studied.

In some classes of work it is a convenient practice to letter each part on the general drawing, and to note the same letters on the specification or order sheet, thus enabling the whole machine to be ordered from the general drawings. This is a very excellent service performed by the general drawing in certain lines of work, but for such a purpose the drawing is quite inapplicable in others.

Merely as a basis for judgment of design, the general drawing fulfils an important function in any class of work, for it approaches the nearest possible to the actual appearance that the machine will have when finished. A good general drawing is, for critical purposes, of as much value to the expert eye of the mechanical engineer as the elaborate and colored sketch of the architect is to the house builder or landscape designer.

From the above it is readily understood that the general drawing, although a mere putting together of parts in illustration, is yet of great assistance in producing finished and exact machine design.

GENERAL COMMENTS ON PRECEDING PROBLEM.

After following through the detail of work as given in the preceding pages, it is worth while to stop for a moment and take a brief survey or review of the subject as illustrated therein.

If the text be carefully studied it will be seen that in every part to be designed the same routine method has been followed, regardless of the final outcome. In some cases it may seem a roundabout procedure to follow a train of thought that finally ends in a design apparently based on purely practical judgment, the theory having had but very little if any influence. The question at once arises—Why not use the empirical rule or formula in the first place? Why not make a good guess at once? Why not save all the time and energy devoted to a careful analysis and theory, if we are finally to throw them away and not base our design on them?

The principle to be noted in this connection is, that it is just as fatal to good design to rely upon bare experience and upon judgment alone, as it is to construct solely according to what pure theory tells us. There are many things in the operation of machinery that are totally inexplicable from the purely practical point of view, and will forever remain so until we analyze them and theorize on them. Many good things in machinery have been the result of what might be called "reversed" machine design. When a new machine is started, it frequently, or we might almost say always, fails to do its work just as it is expected to do it. This is because some little point of design is bad, owing to the inability of drawings, however good they may be, to show all that the machine itself in bodily form and in motion shows.

Now, if our analysis and theory have been good in the designing process, it is almost sure that we can very readily analyze and theorize on the trouble that exists when the machine is finished, can detect the weakness, and can correct it with comparatively small change in the general design. This is "reversed" machine design.

If, on the contrary, we have based our design purely on guesswork, allowing our fancy full and free play to work out the details without further basis, we may consider ourselves lucky if the machine runs at all. This, however, is not the worst of the situation. If the machine does actually operate, even as well as it might reasonably be expected to, but still has the usual difficulty of some little kink or hitch that was not expected, then, as a result of the method upon which the whole thing has been constructed, we have no definite plan of action to proceed upon. We must try first this, then that scheme to obviate the trouble. We may be fortunate enough to "strike it" the first time; we may never strike it. It is doubtful if the machine ever can be made to work at highest efficiency; and if fairly good results be finally obtained we never know the reason why, and have nothing on which to base any future action or design.

This haphazard process is not machine design at all, either in name or in result.

As has previously been stated in these pages, there is no such thing as too much analysis or theory in the designing of machinery. Even if we carefully analyze, theorize with rigorous exactness, and then practically modify our construction to such a point that the original theoretical shape is almost or entirely lost, the apparently roundabout process is not in vain, for we are in perfect control of our design. We know exactly what it has to take in the way of forces, blows and vibrations. We know what its ideal shape should be. We know where we can practically modify its form without weakening it excessively or adding excess of material. In other words we know all about it, and therefore know exactly what we can do with it; and whether it follows in its shape the outline that pure theory gives it or some other outline, it is nevertheless well designed.

"Reversed" machine design, as described above, based on observation and experiment with regard to machines already in operation, is just as impossible without exact analysis and theory as is original design based merely on mechanical ideas in the abstract. The method once learned and made a habit of mind will produce results with equal facility in either case, and results are what the mechanical world is seeking.

Another point worth noting in the progress of the problem as given is the absolute necessity of possessing some knowlege of Mechanics. The more of this subject the designer can have at his finger ends, the more ready and successful will he be in all problems of Machine Design. However, the principles of forces and moments clearly understood, and the application of the same in the all-important subject, "Strength of Beams," constitute a fund of information that will give a splendid start and a good working basis for simple designs. It should always be remembered that a complicated design is little more than a combination of simple designs, and if one has the ability to dissect and analyze what seems at first like a bewildering maze of parts, complication is speedily changed to simplicity.

Common sense goes a long way in good designing. There is nothing mysterious about the process If the beginner will only avoid doing things that are foolish and ridiculous on their very face, if he will exercise the same judgment that he uses in the daily affairs of his life and will mix in something of mechanics and mechanical method, he will be on the direct road to success in the art.

Good drawing is an essential element of good design, and it is especially urged that the sketches and drawings as reproduced in the preceding text be studied with this in mind. By a good drawing is meant not a showy piece of work, finely shaded or artistically lettered, but an exact layout, definite and measurable, correctly dimensioned if in detail, and meaning exactly what it says. Machine design is an exact science, and the designer cannot shirk responsibility by permitting his work to be shiftless and loose. If he cannot delineate clearly and in definite form what he determines in his mind the structure should be, then it is purely good luck if he achieves success, and it may safely be asserted that the success is due to some subsequent care and finished design added to his feeble effort, rather than to any expertness of his own. Such success is of a very doubtful nature, and if not bordering on financial loss it is at least secured only at a low working efficiency.

As examples of good drawings the plates shown are not claimed to be anything extraordinary, but it will be noted that they are clean-cut and definite, and that even the sketches are unmistakable as to that which they are intended to illustrate. The information as to the design is all there; nothing is left to the imagination.

Classification of Machinery. It is intended to be made clear in all that has preceded, that the same method of attack and procedure may be applied to the designing of machinery, whatever may be the class or kind. This is a fundamental principle. When it is logically carried out, however, it produces very different results, as is evidenced by the characteristics of style peculiar to each of the classes of machinery to one or another of which all machines belong.

For example, an engine lathe has a style similar to a drill press, or a boring mill, or a screw machine, or a milling machine. It is very different, however, from the style of a steam engine, or a pump, or an air compressor, or a locomotive; it is still more different from the style of a rolling mill, or a link belt conveyor, or a coal crusher, or a stamp mill.

These classes of machinery are so distinctly marked that the novice is easily able to perceive that there is some controlling influence in each which marks its peculiar style. He should at the same time see that the very analysis that has been so strongly insisted upon in these pages is the direct cause of the marked characteristic in design. Each class of machinery must satisfy certain exacting conditions different from those of any other, and it is the careful study of these conditions, as fundamentally enforced, which leads to the strictly logical design.

A few of the most common classes are enumerated below, and their prominent features noted. It is hoped that a study of them will familiarize the student in a general way with the requirements of each, and serve as a guide to a more comprehensive study of their detail design than is possible in these pages.

Machine Tools. Examples:—lathe, planer, milling machine, drill press, screw machine, boring mill, grinding machine, etc., etc.

The machines of this class are all utilized for the finishing of metal surfaces. They are really at the root of the production of machinery of all other classes. Accuracy is their prime characteristic—accuracy of construction, accuracy of operation, accuracy of adjustment. Any inaccuracy that exists primarily in a machine tool is reproduced in every piece upon which it produces a finished surface ; and since the mere act of finishing a surface upon anything implies that a rough and inaccurate surface will not answer, the tool then fails of its purpose if it cannot produce a true surface: it does not accomplish that for which it was designed.

The effect that this element of accuracy has upon the design of a machine tool is to require long bearings, convenient and exact methods of adjustment, stiffness, excess of material to absorb vibration, special shapes to facilitate application of jigs, fixtures, and exact manufacturing devices insuring interchangeability of parts, dust guards, and automatic lubrication.

Machine tools are essentially machines of maximum output, and depend for their success, not only upon their accuracy as noted, but also upon their ability to do the greatest amount of work per square foot of space occupied, with the least amount of manual labor and attention on the part of the operator. This is especially true of automatic machinery, which perhaps might be classed by itself in this respect, but which is nevertheless included under the broad term of a machine for producing finished surfaces, being merely the highest and most refined form of same. For machines of this class the designer has to study every detail with the most minute attention, packing away the operating parts into the smallest space and yet providing ready means for access, removal, and repair. Clearances that would be too little for other kinds of machinery are permitted and provided for; material of high grade, strength, and wearing quality, though expensive in first cost, and requiring the most expert skill to finish and to fit into place, must be used in order to keep the machine compact and yet of large capacity, to make it reasonably light in weight and yet amply strong.

Another point which has a great influence on the design of a machine tool is that we can never tell in advance just what it will have to stand in work, for the variation in the material that it finishes, the uncertain skill of the operator who runs it, the crowding to its limit of capacity and even beyond in times of press of business, and the many other stresses that may suddenly and without warning be thrown upon it, must all be thought of and provided for.

The points above mentioned are but a few of those which the designer of machine tools has to meet, and are presented merely as illustrations to show the special skill required in this class of machinery. It is readily seen that while the machine tool designer has great latitude in choice of material and in expenditure of money for refinement of structure—perhaps greater latitude than in any other class, yet he is held down as in no other to the final productive results, a small percentage of failure entirely throwing out the machine as a marketable product.

The style and external appearance of machine tools have a character of their own resulting from this extreme detailed care in design. Corners and fillets are carefully rounded; surfaces and intersections are definitely made; in short, the mechanical beauty of a machine tool is seen only from a near view and close inspection, and it is to this end that the design is constantly directed Appearance is a large factor in the sale of a fine tool, and the prestige of the American trade abroad in this respect is very noticeable.

Motive-Power Machinery. Examples:—Steam engine, gas engine, air compressor, steam pump, hydraulic machinery, etc., etc.

The element of heat enters into the design of all machinery in this class. The natural agents, air, gas, and water, in their various forms, are taken into the machine in the most efficient form in which it is possible to obtain them, are robbed of their energy to provide power, and are discharged in a form as weak and inert as the efficiency of the machine will determine.

In contrast to the class of machinery just studied, it should be noted that these machines do not produce any material thing; that is, they do not produce finished surfaces on metals, make screws or bolts, bore holes in castings, or turn line shafting. They merely take the energy of the natural agent, which is not in a form available for use, and transform it into motive power for general use.

Hence the element of accuracy as entering into the design of these machines is necessary only for their own efficient operation, and not for the quality of the thing which they produce, as in the case of machine tools. For example, the power furnished by one steam engine to drive a line shaft is as good as that of another as far as the rotating of the shaft is concerned, provided, of course, that both are equipped with the same quality of governing mechanism. The fact that one of the engines has a good adjusting device on the main bearing while the other has not is of no consequence from the standpoint of the line shaft, but it is, of course, of consequence respecting the efficient operation of the engines.

The design of steam engines and similar machines is of a rough nature compared with that of machine tools, as far as the detail of surface is concerned. General accuracy is nevertheless essential for the machine's own sake, but while in the machine tool we deal with thousandths of an inch, in the steam engine hundredths of an inch indicates fine work.

These machines are subject to extremes of temperature that have to be provided for in the design and arrangement of the parts. Being prime movers, controlling the operation of many machines, they must be certain to run during their period of work; hence design and adjustment must be positive, and when the latter cannot be made while running, it must be quickly and definitely accomplished when a stop is made. Simplicity of construction is essential, facilitating cheap and quick repairs. The design should be such that constant attention while running is avoided, the usual attention of the engineer being a safeguard rather than an implied factor of the original design. General rigidity and stiffness are important, also good balancing of the moving parts, and weight for absorption of vibration; otherwise under the constant daily run the machines will tear to pieces not only themselves but their foundations.

As far as external appearance goes in this and subsequent classes to be mentioned we are on a very different basis from that of machine tools. *General* mechanical symmetry of form is aimed at in the design, and the several smaller parts depend for their outline (aside from considerations of strength, which are, of course, always in order) upon the harmonious relation which they bear to the main and fundamental elements of the machine. Such machinery as air compressors, steam engines, pumps, and the like are viewed as a whole, and criticised, not detail by detail, as is the machine tool, but as to general effect of outline observed from some distance. To convey the desired effect to the eye the design must be bold and massive, connections simple and direct, and the smaller parts must not be so dwarfed in size as to appear like delicate ornaments instead of integral parts of the machine. The lines of connected parts must be continuous from one part to the other; and when interrupted by flanges, bosses, or lugs, the latter, which are merely incidental to the former must not be allowed to obscure wholly the main lines of the fundamental pieces.

It is attention to such points as these that marks the difference between well-designed motive-power machinery and that of the opposite character. Even though the little details of fillets and corners and surfaces may have their effect from a close point of view, the design will stand or fall in excellence on its bolder features, as noted above.

Structural Machinery. Examples:—Hoists, cranes, elevators, transfer tables, locomotives, cars, conveyors, cable-ways, etc., etc.

In the two preceding classes that have been noted, cast iron in the form of foundry castings enters as the principal material. Steel is utilized for shafts, studs, pins, and keys. Also special forgings, malleable iron and steel castings enter as factors in the production of the machinery discussed. Foundry castings, however, compose the great body of the material used, and the chief problems involved are those of the expert moulding of cast iron, and the handling and finishing of the same. For the operating parts, steel of fine grade is used in highly finished form, expensive because of its fineness, and yet a necessity to the extent it is used. Brass and bronze are used in the same way, generally in connection with the bearings for the shafts.

Structural machinery, on the contrary, uses steel as the basis of its construction. The fundamental structure is built up of plates, channels, beams, and angles; castings, though numerous, are relatively small, being riveted or bolted to the main structure and controlled in their design by its requirements.

Steel is used in this manner partly because the exclusive use of castings is prohibited on account of the excessive weight, and therefore expense, and partly because castings could not be made which would possess the necessary toughness and strength. In many cases the size of the machinery is such that castings, even if they could be made, would not support their own weight. Moreover, machinery of this class is subjected to rough service, and yet must be practically infallible under all conditions, neither being uncertain in operation at critical moments nor entirely failing under an extraordinary load.

The design of structural machinery is tied up to conditions existing largely outside of the locality in which the machinery is built. The steel plates and structural shapes required, being products of the rolling mill, have to conform to the latter's standards. The rivets, bolts and other fastenings have to be in accordance with the established practice of the structural iron worker, in order to permit punching, shearing and bending machinery of regular form to be utilized. Shipment on standard railway cars has to be considered, the design often requiring to be modified to permit this and nevertheless insure positive and accurate assembling in the field.

Steel castings, both large and small, find ready application in this class of work; also steel forgings, requiring to be worked under a heavy hammer and in many cases by specially devised processes.

In structural design less of the actual process of manufacture is under the eye of the designer than in the former classes of machinery which have been considered, and hence more allowance has to be made for things not coming exactly right to the fraction of an inch. It would be bad design to plan any structural piece of work with the same closeness of detail permitted, and in fact required, in the case of machine tools, or even in the case of motive-power machinery. In planning structural work the idea must be carried out, of certainty of operation in spite of roughness of detail and variations of construction. This does not necessarily imply inaccuracy, or shiftless, loosely constructed machinery; on the contrary, quite the reverse. The locomotive, for example, is one of the most refined pieces of mechanism that exists today; and yet the methods applied to the construction of machine tools would prove a failure on the locomotive. The design of a car axle box has to be just right else it will heat and destroy itself; the same is true of the spindle of a fine engine lathe; and yet how rough the former is compared with the latter, and how unsuited either would be for use on the service of the other.

As a general rule structural. machinery can be more closely proportioned to theoretically calculated size than can the preceding types. The rolled material of which it is made is of a uniform and homogeneous nature owing to its process of manufacture, hence its every fibre may be counted on to sustain its share of the total load imposed upon it. This is in sharp contrast to the case of cast iron, which is of such a porous and irregular structure that we have to use a large factor of safety to cover this inherent defect.

Steel castings of both small and large size (which are quite apt to be utilized in this class of machinery for parts that can with difficulty be made out of rolled material), if properly designed of uniform thickness, with all corners well filleted and with the channels for the flow of the molten metal direct and ample, are nearly as reliable as rolled steel. In parts subject to excessive vibration, shocks, and sudden wrenchings, as, for example, the side frames or the connecting rod of a locomotive, the forged and hammered material is practically a necessity. This is especially the case when the possible breakage of the part would cause serious consequences involving heavy loss of life and property.

From the several points of view as above considered, it can be readily appreciated that, while structural work is in one sense rough and unpolished, yet it requires, from an engineering standpoint, quite as much breadth of experience and judgment as any of the other types. The fine-tool designer, least of all, perhaps, requires book theory, but does require an extended machine-shop experience. The designer of motive-power machinery needs pure physical theory and shop experience of a large and broad scope. The structural designer is least of all concerned with refined and minute finishing processes, but utilizes his theory absolutely, even though roughly.

Mill and Plant Machinery. Examples:—Rolling mills, mining machinery, crushers, stamps, rock drills, coal cutters, the machinery of blast furnaces and steel mills, tube mills, etc., etc.

This machinery constitutes a class which in the roughness of its operation exceeds all others. Moreover, it is machinery which for the most part is in continuous operation—24 hours per day and 365 days in the year. Hence refinement, even such as might be permitted in the preceding class of Structural Machinery, would be fatal here. The conditions that surround plant machinery are unfavorable in the extreme to the life of any material or metal, and it is not possible to change these conditions or give more than partial protection to the operating parts. Hence the design of such machinery must proceed primarily on the assumption that abuse and neglect, grinding away of surfaces, chemical eating away of metal, flooding of parts with water gritty and corrosive, subjection to sudden bursts of flame and intense heat, etc., will in a relatively short time totally destroy, perhaps, the entire structure.

In view of the continuous nature of the working process, which must be kept up in spite of these almost insurmountable conditions, the problem in each case becomes one of expediency; and the designs and arrangement of machinery must be so worked out that operation, repair, construction, and installation can all go on simultaneously without stopping the continuous process, and with but a small degree of inconvenience to the operation of the plant.

This problem, difficult though it may seem, can be worked out successfully, as is evidenced by the great number of plants of the continuous character operating at high efficiency throughout the world. The engineering and designing skill required to accomplish this, is perhaps of the highest degree met with in modern practice, for in it is involved a working knowledge of the possibilities, if not the detailed designs of machinery included in all classes. And yet, as in the most elementary case of simple design that can be conceived, the result is accomplished in the same way, namely, by studying the conditions (analysis), developing an ideal application to those conditions (theory), and then reducing the ideal design to a practical basis (modification).

A Few Pointed Suggestions on Original Design. Original design deals with the development of original mechanical ideas. The prime requisite for the development of an idea is to understand thoroughly the idea in the rough. See distinctly the mark aimed at, and never lose sight of it. If a method of reaching it is already outlined, understand that also thoroughly and the principles involved. It is impossible to go ahead blindly and hope to come out right. No good machine was ever built that does not stand for hours of concentrated thought on the part of its designer. Good machines never *happen*, they always *grow*.

Just as soon as the object to be accomplished is clearly understood, begin to produce some visible work on the problem. Sketch something. Get some ideas on paper. Ideas on paper suggest other ideas. If the problem, for example, is one of lathe design, sketch a rectangle, and call it the headstock; another rectangle, and call it the footstock; a couple of scratches for the centers; some steps for the cone pulley; three or four lines for the bed; and as many more for the supports. There is now something on paper to look at; the design is begun.

It is much better to stare at this sketch, than into blank space trying to imagine the finished design. No matter how rough the sketch may be, a short study of it will develop some limiting conditions that before were not apparent. Guess at a few rough dimensions; put them on the sketch; develop another view—a plan or a side elevation—all still in the roughest style, without any regard to finished detail. Information will be growing all the while, and the problem will be opening up. At this stage it is probable that the sketch can easily be seen to be wrong in many respects. Perhaps the arrangement will not do at all.

This is a good sign. It shows that the design is progressing. It is a valuable thing to know that certain plans *cannot* be followed. Do not rub out part of the sketch already made and try to remedy it. Begin again. Make another sketch. Sketch paper is cheap. By and by it may prove to be very desirable to have that first rough outline available for comparison; or it may be that some of its ideas can be applied on other sketches. The second sketch may "show up" little or no better than the first. Make another, and another, and another, until the subject is thoroughly digested. It is wonderful how helpful it is to have some marks on paper relative to a design, even though they be of the utmost crudeness. They save imaginative power tremendously; and, even with them, all available powers of imagination will be needed before the design is perfected.

A careful comparison of one's sketches, rejecting here, and approving there, will, little by little, bring about a definite opinion, and the scale drawing can be begun.

As in the case of the first sketch, so in the case of the first scale drawing, get some lines on paper as quickly as possible. Draw something, even if it is nothing more than a straight horizontal line. Do not stare at blank paper for an hour trying to imagine how the tenth or eleventh line is going to be drawn in relation to the first line. Do not worry about the later lines until it is time for them Draw the first line at once; and, when the second line is drawn, if the first line proves to be wrong, make it right. As in the rough sketch, that first horizontal line is an immense relief from the great waste of blank paper of a fresh sheet. It is something to look at. It is the beginning of a detailed design. If it happens not to be the absolutely correct foundation to build upon, it at least is something to tear down. The main purpose of these preliminary drawings is to keep the mind active on the problem; and advance toward the final accomplishment of the design is often made quite as rapidly by discovering what to tear down as by consistently building up.

When a detail draftsman who has been used to having all his work laid out for him by an expert designer attempts to take up original work for himself, he encounters the drawing of that first line in a way he never did before. He is apt to worry for some time over the possible or impossible results of drawing that first line. If he continue this, he will be sure to fail. The second line is much easier to draw than the first, and the third than the second; and the next hundred will follow on in comparatively smooth sequence, all because of bold action on the first few lines.

And yet, just as the design appears to be progressing smoothly, and the advanced progress of the drawing seems cause for congratulation, careful consideration may disclose a "snag" not previously known to exist in the problem. Further study pursued along the line of this new discovery may show that the whole layout thus far has been radically wrong, and that a fresh start will have to be made. At such a time the young designer is apt to feel that his labor has all been thrown away, and he becomes discouraged. There is, however, no cause for discouragement. Machine Design might almost be defined to be the "successful elimination of snags." It takes some ability to discover an obstacle of this sort; to know a "snag" when an opportunity to see it is given. It takes a good designer to eliminate such a difficulty after it has been found. If there were no "snags" it would not require great ability to design machines. Many machines fail because in them there are a lot of undiscovered "snags." Others fail because the "snags," although discovered, were not eliminated by careful design.

Do not be afraid to make a lot of "first" drawings. It is just as important to digest the design thoroughly by means of scale drawings, as it was to digest it originally by means of the rough sketches. An attempt to make the first drawing of an original design absolutely right would, it is safe to say, produce a poor design, one that could be much improved by further trial. Let the drawings multiply, one after another, until the final one is reached, in which the perfection of detail will eliminate all the bad points of the preceding drafts and incorporate good ones of its own based on the study of the others.

And yet it is often true that the first design laid out, even after many others have been developed, may be found to possess features that render a return to it desirable. This is why it is always better to produce a collection of designs than to attempt to rub out and work over the first one. The best designers usually have a great number of sketches showing how to accomplish a single result. Likewise, they also have a series of layouts to scale, showing in detailed form the development of their various ideas. This is because, without a careful consideration of many methods, they themselves feel incompetent to judge of the best design possible for accomplishing a given result.

Sketches and original designs should always be dated and signed. Different designers may be working on the same problem, and priority of design will never be allowed except upon signed and witnessed papers. It is embarrassing to find, after months and perhaps years have passed since an original drawing was made, that one's rights have been preëmpted merely because there was no date or signature to define them.

In redesigning or modifying an existing machine, never make a change merely for the sake of doing so. Give the good points of the machine a chance, and devote attention in the new design to correcting the bad points. It is in bad taste, if it be not actually childish, to "look wise and suggest a change" in details which happen to have been designed by another party, but which, nevertheless, are by common engineering judgment pronounced good for the special work intended. This element of unfair and selfish criticism has more than a moral bearing. When it is carried into the superintendence of designing work, it extinguishes the personality of the subordinate draftsman; his efficiency as an original thinker is lowered; and narrow designs are produced.

"The best way for a subordinate to dispose of what appears to be a poor suggestion from a superior, is to work it out to the best degree possible." If it turns out to be good the credit of working it out belongs to the man who did it. If it is actually bad, a careful working out will usually develop the fact beyond dispute, and save unprofitable argument. For the success or failure of a machine there is only one argument better than the detail drawings, and that is the machine itself in operation.

Detail drawings, however, are infinitely better prosecutors or defendants than a multitude of wordy counsel.

Summary. The above classification of machinery might be subdivided and extended indefinitely, and on the broad basis on which it is given it doubtless does not cover the entire field. As an illustration, however, not only of types of machinery, but of methods of design and study, it is hoped that it may be of assistance in giving a start to the student of machine design, in whatever class his interests may happen to lie.

It is the general principles of the art which it is important to master. It is not the designing of a locomotive, or a stationary steam engine, or a crane, or an engine lathe, or a rolling mill, which should be sought to be learned, but the designing of *anything* that may confront us. Specializing is sure to come to the designer in the course of his experience, and when it does he merely fits to the particular specialty the principles he knows for all, and practically develops them along that individual line.

EXAMINATION PAPER

MACHINE DESIGN.

The drawings made in accordance with the problems below should be traced in ink on tracing cloth 18 by 24 inches in size, and having a border line $\frac{1}{2}$ inch inside the edge of the paper

PROBLEMS.

1. Suppose a 30-inch pulley is substituted for the 42-inch in the problem given, and that the pulley on the motor remains $10\frac{1}{2}$ inches as before, how fast must the motor run to give the rope the same speed, 150 feet per minute ?

2. Will the horse-power of the motor be changed with this new condition ? Explain fully.

3. Calculate the width of double belt for above condition.

4. What is the torque on the motor shaft for above condition?

5. Calculate the size of shaft in the small pulley for above condition.

6. Calculate the size of shaft in the 30-inch pulley for above condition.

7. Design and draw both pulleys for above condition, making complete working drawings, and giving all calculations in full.

8. Taking the original problem as given in the text, suppose it is desired to increase the large gear to 45 inches diameter, calculate the load on the tooth, and a suitable pitch and face to take this load.

9. How many teeth must the pinion have to give the same speed of rope, 150 feet per minute, assuming that the motor runs 470 revolutions per minute, for condition in Problem 8 ?

10. Calculate the bore of pinion for this case.

11. Design and draw the gears for the conditions of Problems 8 and 9, giving all calculations in full.

12. When there is but 3,000 pounds on the rope, what are the tensions in each end of the brake strap, assuming that the size of drum and other conditions remain the same?

13. How much pressure on the foot lever would it take to hold this load of 3,000 pounds on the rope?

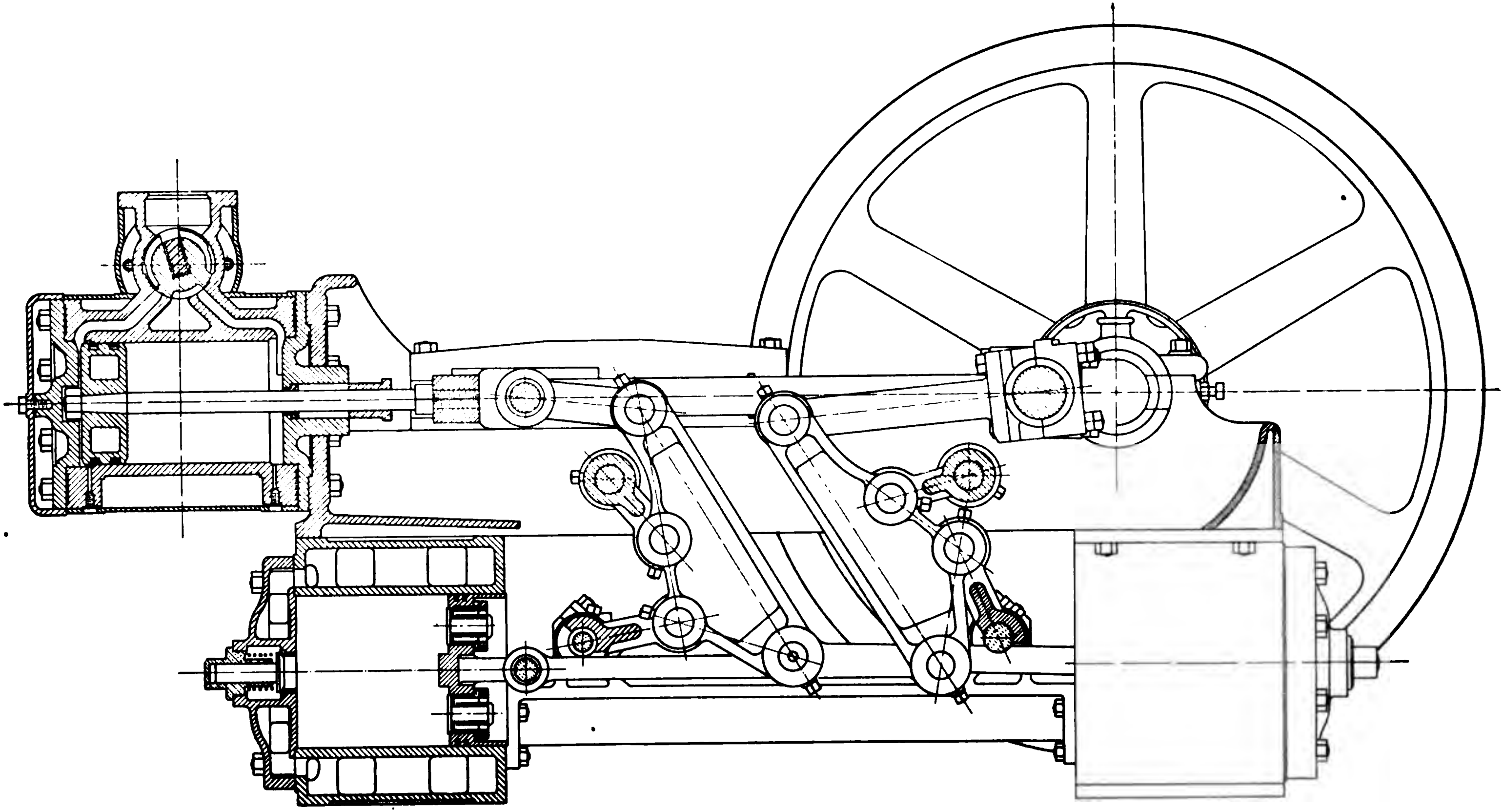
14. Suppose we put a bearing 9 inches long on the drum shaft; the distance, center to center of bearings, would then be 3 feet 8¾ inches, gears, drum, brake, and load being same as in the original problem of the text. Calculate the diameter of the drum shaft.

15. Suppose the height of bracket, center to base, to be 15 inches; length and diameter of bearing, as in Problem 14; and that we use a separate bracket for the drum bearings, not connected with the pinion-shaft bearings. Design and draw such a bracket.

16. Calculate, design, and draw all the parts for a machine similar to that of the text, from the following data:

Load on rope....................4,000 pounds.
Speed of rope....................175 feet per minute.
Length of rope to be reeled in....250 feet.

Problem 16, is supposed to be worked out on the same lines as the text, but is wholly original in its nature, being based on entirely new data. It is not expected that this problem will be attempted except by well-advanced students who can give considerable time to working it out completely. It will be found, however, an excellent exercise in original and yet simple design.



AIR COMPRESSOR (N. Y. AIR BRAKE CO.)

General or Assembled Drawing

MACHINE DESIGN.

PART II.

Introduction. In Part I is illustrated a definite and systematic method of attacking the design of a *machine as a whole*. In Part II the same plan is followed with regard to the detail of its *component parts*, the machine elements which are chosen as illustrations of the method, being the simplest and most familiar forms in common use.

As before, the student must strive to grasp and absorb the *method* of design rather than any specific and established form of a machine part. Part II is not a compendium of design, does not attempt to be complete or exhaustive in any of its chapters, but is condensed and simplified in order to lead the student into systematic mechanical thinking and logical and definite action. Each chapter is intended to stimulate to further and more exhaustive study along lines broader than, and under conditions different from those that can be specified in a general discussion. But no matter how deeply investigation may be carried, or how specialized the study may become, the student must realize that his path of action in any case whatsoever must lie along the lines of *Analysis*, *Theory*, and *Practical Modification* systematically applied.

BELTS.

NOTATION—The following notation is used throughout the chapter on Belts:

A = Sectional area of belt (square inches) $= bh$.
b = Width of belt (inches).
F = Force of friction at pulley rim (lbs.).
h = Thickness of belt (inches)
μ = Coefficient of friction.
N = Number of revolutions of pulley per minute.
n = Fraction of circumference of pulley embraced by belt.
P = Driving force at pulley rim (lbs.) $= F$.
R = Radius of pulley (feet).
r = Radius of pulley (inches).
T = Initial tension (lbs.).
T_n = Total tension on tight side (lbs.).
T_o = Total tension on slack side (lbs.).
t = Working tension of belt (lbs. per square inch).
V = Velocity of belt (feet per minute).
w = Weight of belt per cubic inch (lbs.).
z = Factor due to centrifugal force.

ANALYSIS. When a belt is stretched over a pair of pulleys, is cut off at the proper length, and is laced together into an endless band, it is evident that as long as the belt is at rest there is a nearly uniform tension in it throughout its length, due to the tightness with which the lacing is drawn up. If the distance between the pulleys is considerable, the weight of the belt itself as it hangs between the pulleys will produce a slightly greater tension next to the pulleys than exists in the middle of the span. This increase of tension due to the weight of the belt would make but little difference in the unit-stress in the material of which the belt is made; hence it may safely be assumed that the tension in the belt when at rest is uniform throughout its entire length.

When we start to transmit power through the belt by turning one of the pulleys, thereby driving the other pulley the condition of stress in the belt is at once materially changed. As the belt is a flexible member, we can transmit only a pull to the other pulley, thereby turning it around, the push which is at the same time given to the other side of the belt merely acting to make the belt sag or become slack. Hence the immediate effect of starting motion in a belt is to change the condition of equal tension throughout its length, to that of unequal tension in the two sides. The driving side is tight, while the other is loose, the former having gained as much tension as the latter has lost, and the sum of the two being practically equal to the sum of the tensions in the two sides of the belt when at rest. This is not strictly true, as will be shown later; but it is sufficiently accurate to form a good basis for the practical design, at least of slow-speed belts.

This condition of tight and slack sides is made possible by the fact that the belt, in being wrapped around the pulleys under tension, has friction on their surfaces. Thus, we can pull hard on one side without slipping the belt around the pulleys, but could not do this if the pulleys were perfectly smooth or frictionless, for in that case the slightest pull on one side would slip the belt around the pulleys. In fact, it would be impossible to produce any pull by means of the driving pulley, for the pulley would merely slip around inside the belt.

The amount of pull we can apply to the belt is therefore limited by the tension at which the belt slips around the pulley. Moreover, since the force of friction between the belt and pulley is dependent upon the normal force with which the belt is pressed against the pulley, and the coefficient of friction between the two, it is evident that the tighter the belt is laced up, and the rougher the surfaces of the pulley and belt, the greater is the force that can be transmitted through the belt. This leads to the conclusion that it would be possible to transmit any amount of power through any belt however small, if the belt were only laced up tight enough.

This conclusion is literally true; but the important fact now comes in, that the strength of the material of which the belt is made is limited, and while theoretically we might be able to accomplish the above, it would be impossible to do so in practice, for at a certain point the belt would break under the strain. Other practical considerations also come in, which fix this limit of power transmission at a point far below the breaking strength of the material.

The complete analysis is not quite as simple as the above, especially for high-speed belts. When the driving side of the belt becomes tight, it stretches and grows longer; and at the same time the other side of the belt becomes slack and grows shorter. But it is not true that the increase in the one side is the same as the decrease in the other, and this fact produces the condition that the sum of the tensions in motion is not quite the same as the sum of the tensions at rest.

Again, when the belt, as it passes around the pulley, changes its straight-line direction to circular motion, each particle of the belt—like a body whirling at the end of a cord about a center of rotation—tends by centrifugal force to fly away from the surface of the pulley, thereby decreasing the normal pressure, and hence the friction. This centrifugal force also changes somewhat the tensions in the belt between the pulleys. As the centrifugal force increases in proportion to the square of the linear velocity, it is evident that the effect is greater at high speeds than at moderate or low speeds.

A further circumstance that affects the driving power of a belt is the stiffness of the leather or other material of which the belt is made. As it passes around the pulley, the belt is bent to conform to the circumference of the pulley, and is again straightened out as it leaves the pulley. Hence the theoretically perfect action is modified somewhat according to the sharpness of the bending and the thickness or flexibility of the belt; in other words, a small pulley carrying a thick belt would be the worst case for successful calculation on a theoretical basis.

THEORY. The condition of the tight and loose sides of a belt transmitting power, is similar to that of the weighted strap and fixed pulley shown in Fig. 17. If motion is desired of the strap around the pulley, it is necessary to make the weight W_2 of such a magnitude that it will overcome not only the weight W_1, but also the friction between the strap and the pulley. The strap tension T_n is, of course, equal to W_2, and T_0 to W_1. The equation showing the balance of forces for the condition when motion is about to occur, is:

$$T_n - T_0 = F = P \text{ (driving force).} \qquad (5)$$

If the pulley be free to turn on its axis, instead of being fixed as in Fig. 17, the strap by its friction on the pulley will turn the pulley, and the force of friction F becomes the driving force for the pulley as noted in equation 5 above.

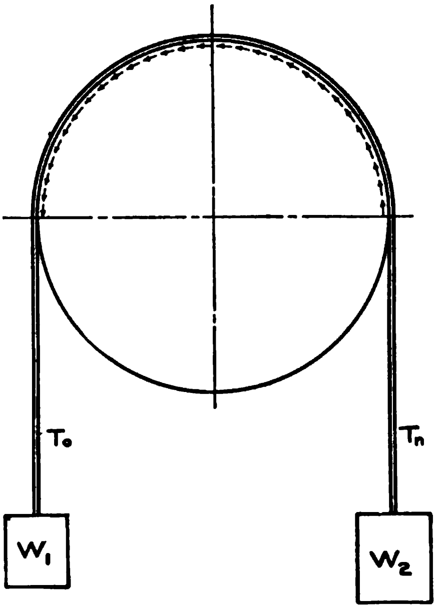


Fig. 17.

In Fig. 18, let us suppose that W is a weight representing the resistance to be overcome. The tensions T_n and T_0, equal at first owing to stretching the belt tightly over the pulleys at rest, change when an attempt is made to raise the weight by turning the larger pulley; and just as the weight leaves the floor, the equality of moments about the axis of the driven pulley gives the following equation:

$$(T_n - T_0)\,r = F \times r = P \times r = W \times r_1. \qquad (6)$$

This equality of moments remains as long as the motion of the weight is uniform, and represents closely the conditions under which belt pulleys work.

Although we know from the above what the *difference* of the belt tensions is, and what this difference will do when applied to the surface of a given pulley, we do not yet know what either T_n or T_o actually is; and until we do know, we cannot correctly proportion the belt. Hence we must find another relation between T_n and T_o which we can combine with equations 5 and 6. This relation is deduced by a process of higher mathematics, which results as follows:

$$\text{Common logarithm } \frac{T_n}{T_o} = 2.729\,\mu\,(1 - z)n. \qquad (7)$$

Treating equations 5 and 7 as simultaneous, values of both T_n and T_o can be found by the regular algebraic solution. As T_n is the larger, the actual area of belt to provide the necessary strength must be made to depend upon it.

The factor z in equation 7 depends upon the centrifugal force developed by the weight of the belt passing around the pulley. Its value, found from mechanics, is:

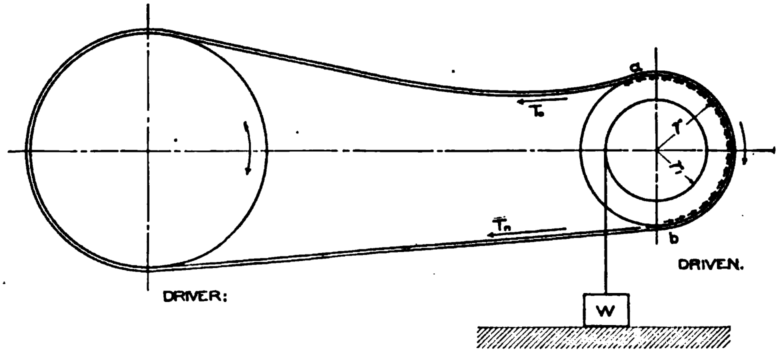


Fig. 18.

$$z = \frac{w \times V^2}{9{,}660 \times t}.$$

Having found the maximum pull on the belt, it now remains to write the equation:

$$\text{External force} = \text{Internal resistance};$$

or,
$$T_n = b \times h \times t. \qquad (8)$$

Usually the most convenient way to handle this equation is to assume h and t, and then solve for b.

Summing up the theoretical treatment of belt design, we simply combine equations 5, 6, 7, and 8, and solve for the quantity desired. Discussion of the constants involved in these equations, and of the practical factors controlling them, is given in the following:

PRACTICAL MODIFICATION. The force of friction F, which is the same as driving force P, depends on:

Coefficient of friction (μ) between belt and pulley;

Tightness of the belt;

Centrifugal force of the belt;

Angle of contact of belt with pulley.

The coefficient of friction (μ), according to experiments and observed operation of belts transmitting power, varies from .15 to .56 for leather on cast iron. An average value consistent with a reasonable amount of slip, the belt being in good running order, is .30. If the belt is oily, or likely to become so in use, a lower value should be taken.

The tighter the belt is drawn up, the greater is the pressure against the pulley, and hence the greater is the force of friction. But if we pull the belt up too tightly, when we begin to drive, T_n becomes too great, and the belt breaks or is under such stress that it wears out quickly. Moreover, the great side pressure on the bearings carrying the shaft produces excessive friction, and the drive is inefficient. This is why a narrow belt driven at high speed is more efficient than a wide belt at slow speed, for we cannot pull up the former as tightly as the latter without overstraining it, and yet it is possible to get the required power out of the narrow belt by running it at high speed.

The centrifugal force is of small importance for low speeds, say of 3,000 feet per minute and less; and it therefore may usually be neglected. The factor z then becomes zero in the expression $1 - z$ in equation 7, and the second member of the equation stands simply $2.729 \times \mu \times n$.

The angle of contact of belt with pulley is important, as a large value gives a great difference between T_n and T_0; and it is desirable to make this difference as great as possible, because thereby the driving force is increased. The loose side of a horizontal belt should always be above, as then the natural sag of the loose side due to its slackness tends to increase the angle of contact with the pulley, while the tightening up of the lower side acts against its sag to make the loss of wrap as little as possible. Vertical belts which have the driving pulley uppermost, utilize the weight of the belt to increase the pressure against the surface of the pulley, slightly increasing its capacity for driving. The angle of contact may be artificially increased by a tightening pulley which presses the belt further around the pulley than it would naturally lie. It adds however, the friction of its own bearing, and impairs the efficiency of the drive. For ordinary horizontal belts, the angle of contact is but little more than 180°, and the value of n in equation 7 may be safely assumed at $\frac{1}{2}$ unless the pulleys are of relatively great difference of diameter and very close together.

Strength of Leather Belting. The breaking tensile strength of leather belting varies from 3,000 to 5,000 pounds per square inch. Joints are made by lacing, by metal fasteners, or by cementing. The strength of a laced joint may be about $\frac{7}{10}$, of a metal-fastened joint, about $\frac{1}{2}$, and of a cemented joint, about equal to the full strength of the belt cross-sectional area. The proper working strength of belting depends on the use to which the belt is put. A continuously running belt should have a low tension in order to have long life and a minimum loss of time for repairs. For double leather belting it has been shown that a working tension of 240 pounds per square inch of sectional area gives an annual cost — for repairs, maintenance, and renewals — of 14 per cent of first cost. At 400 pounds working tension, the annual expense becomes 37 per cent of first cost. These results apply to belts running continuously; larger values may be used where the full load comes on but a short time, as in the case of dynamos.

Good average values for working tensions of leather belts are:

Cemented joints,	400	pounds	per	square	inch.
Laced joints,	300	"	"	"	"
Metal joints,	250	"	"	"	"

Horse-Power Transmitted by Belting. If P is the driving force in pounds at the rim of the pulley, and V is the velocity of the belt in feet per minute, the theoretical horse-power transmitted is evidently :

$$\text{H. P.} = \frac{P \times V}{33{,}000}. \qquad (9)$$

It is evident from the above that the horse-power of a belt depends upon two things, the driving force P and the velocity V. If either of these factors is increased, the horse-power is increased. Increasing P means a tight belt. Hence a tight belt and high speed together give maximum horse-power. But a tight belt means more side strain on shaft and journal. Therefore, from the standpoint of efficiency, *use a narrow belt under low tension at as high a speed as possible.*

Empirical rules for horse-power of belting, if used with judgment, give safe results when applied to very general cases. A common rule used by American engineers is:

$$\text{H. P.} = \frac{b \times V}{1{,}000}. \qquad (10)$$

For a double belt, assuming double strength, this becomes:

$$\text{H. P.} = \frac{b \times V}{500}. \qquad (11)$$

With large pulleys and moderate velocities, this may hold good. With small pulleys and high velocities, however, the uncertain stresses induced by the bending of the fibers of the belt around the pulley, and the relatively great loss due to centrifugal force, modify this relation· and a safer value for a double belt of the ordinary kind is:

$$\text{H. P.} = \frac{b \times V}{540}; \qquad (12)$$

or, still safer, $$\text{H. P.} = \frac{b \times V}{700}. \qquad (13)$$

If we compare the theoretical value of equation 9 with the empirical value of equation 10 by putting them equal to each other, thus:

$$\text{H. P.} = \frac{P \times V}{33{,}000} = \frac{b \times V}{1{,}000}.$$

and solve for P, we get :

$$P = 33b. \qquad (14)$$

This develops the fact that the empirical rule of equation 10 assumes a driving force of 33 pounds per inch of width of single belt.

Another way of expressing equation 10 is: A single belt will transmit one horse-power for every inch of width at a belt speed of 1,000 feet per minute.

Speed of Belting. The most economical speed is somewhere between 4,000 and 5,000 feet per minute. Above these values the life of the belt is shortened; also "flapping," "chasing," and centrifugal force cause considerable loss of power. The limit of speed with cast-iron pulleys is fixed at the safe limit for bursting of the rim, which may be taken at one mile per minute.

Material of Belting. Oak-tanned leather, made from the part of the hide which covers the back of the ox, gives the best results for leather belting. The thickness of the leather varies from .18 to .25 inch. It weighs from .03 to .04 pound per cubic inch. The average thickness of double leather belts may be taken as .33 inch, although a variation in thickness from $\frac{1}{4}$ inch to $\frac{7}{16}$ inch is not uncommon. Double leather belts may be ordered light, medium, or heavy.

In a single-thickness belt the grain or hair side should be next to the pulley, for the flesh side is the stronger and is therefore better able to resist the tensile stress due to bending set up where the belt makes and leaves contact with the pulley face. Double leather belts are made by cementing the flesh sides of two thicknesses of belt together, leaving the grain side exposed to surface wear.

Raw hide and semi-raw hide belts have a slightly higher coefficient of friction than ordinary tanned belts. They are useful in damp places. The strength of these belts is about one and one-half times that of tanned leather.

Cotton, cotton-leather, rubber, and leather link belting are some of the forms on the market, each of which is especially adapted to certain uses. For their weights and their tensile and working strengths consult the manufacturers' catalogues.

A prominent manufacturer's practice in regard to the sizes of leather belting will be found useful for comparison, and is indicated in the table on page 12.

Initial Tension in Belt. On the assumption that the sum of the tensions is unchanged, whether the belt be at rest or driving, we should have the following relation :

$$T_n + T_o = 2T;$$

whence,

$$T = \frac{T_n + T_o}{2}. \qquad (15)$$

This is not strictly true, however, as is stated in the "Analysis" of "Belts." It has been found that in a horizontal belt working at about 400 lbs. tension per square inch on the tight side, and having 2 per cent slip on cast-iron pulleys (*i. e.*, the surface of the driven pulley moving 2 per cent slower than that of the driver), the increase of the sum of the tensions when in motion over the sum of the tensions at rest, may be taken at about $\frac{1}{3}$ the value of the tensions at rest. Expressing this in the form of an equation

Sizes of Leather Belting.

WIDTH.	THICKNESS.	
	Singlo.	Double.
1 inch.	$\frac{5}{32}$ inch.	$\frac{5}{16}$ inch.
2 "	$\frac{3}{16}$ "	$\frac{5}{16}$ "
3 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
4 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
5 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
6 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
10 "	$\frac{5}{16}$ "	$\frac{3}{8}$ "
12 "	....	$\frac{3}{8}$ "
14 "	....	$\frac{13}{32}$ "
20 "	....	$\frac{7}{16}$ "

$$T_n + T_o = \frac{4}{3}(2T) = \frac{8 \times T}{3}.$$

$$T = \frac{3}{8}(T_n + T_o). \qquad (16)$$

The value of T thus found would be the pounds initial tension to which the belt should be pulled up when being laced, in order to produce T_n and T_0 when driving.

This value is not of very great practical importance, as the proper tightness of belt is usually secured by trial, by tightening pulleys, by pulley adjustment (as in motor drives), or by shortening the belt from time to time as needed. It is worth noting, however, that for the most economical life of the belt it would be very desirable in every case to weigh the tension by a spring balance when giving the belt its initial tension. This, however, is not always easy or even feasible; hence it is a refinement with which good practice usually dispenses, except in the case of large and heavy belts.

PROBLEMS ON BELTS.

1. Determine the belt tensions in a laced belt transmitting 50 horse-power at a velocity of 3,500 feet per minute. Suppose that the arc of contact is 180°; weight of belt = .035 pound per cub. in.; and coefficient of friction 25 per cent.

2. What is the width of above belt if it is $\frac{3}{16}$ inch in thickness?

3. What initial tension must be placed on above belt?

4. The main drive pulley of a 120-horse-power water wheel is 6 feet in diameter. A cemented leather belt is to connect the main pulley to a 3-foot pulley on the line shafting in a mill. The horizontal distance between centers of shafting is 24 feet; coefficient of friction, 30 per cent; revolutions per minute of line shafting, 180. Design the belt for this drive.

5. An 8-inch double belt $\frac{3}{8}$ inch thick connects 2 pulleys of 30-inch and 20-inch diameter respectively. The horizontal distance between the centers is 12.5 feet. The coefficient of friction is 0.3, and the weight of belt per cubic inch is 0.035 pound. Working tension, 300 pounds per square inch. Speed of belt 5,000 feet per minute. Lower face of 30-inch pulley is the driving face. Required the H. P. which may be transmitted (theoretically).

6. Compare the theoretical horse-power in problem 5 with that obtained by the use of empirical formula.

PULLEYS.

NOTATION—The following notation is used throughout the chapter on Pulleys:

A = Area of rim (sq. in.).
a = " " arm (" ").
b = Center of pulley to center of belt (inches; practically equal to R).
C_1 = Total centrifugal force of rim (lbs.).
c = Distance from neutral axis to outer fiber (inches).
D = Diameter of pulley (inches).
D_1 = " " hub (").
d_1 = " " bolt at root of thread (inches).
d = Diameter of bolt holes (inches).
g = Acceleration due to gravity (ft. per sec.).
h = Width of arm at any section (inches).
I = Moment of inertia.
L = Length of arm, center of belt to hub (inches).
L_1 = Length of rim flange of split pulley (inches).
l = Length of hub (inches).
N = Number of arms.
n = " " rim bolts, each side.
P = Driving force of belt (lbs.).
P_1 = Force at circumference of shaft (lbs.).
P_2 = Force at circumference of hub (lbs.).
p = Stress in rim due to centrifugal force (lbs. per sq. in.).
R = Radius of pulley (inches).
S = Fiber stress (lbs. per sq. in.).
s = Fiber stress in flange (lbs. per sq. in.).
T = Thickness of web (inches).
t = " " rim (").
t_2 = " " " bolt flange (inches).
T_n = Tension of belt on tight side (lbs.).
T_0 = " " " " loose " (").
v = Velocity of rim (ft. per sec.).
w = Weight of material (lbs. per cub. in.).

ANALYSIS. If a flexible band be wrapped *completely* about a pulley, and a heavy stress be put upon each end of the band, the rim of the pulley will tend to collapse just like a boiler tube with steam pressure on the outside of it. A compressive stress is induced which is very nearly evenly distributed over the cross-section of the rim, except at points where the arms are connected thereto. At these points the arms, acting like rigid posts, take this compressive stress. Now, a pulley never has a belt wrapped *completely* round it, the fraction of the circumference embraced by the belt being usually about $\frac{1}{2}$, and seldom, even with a tightener pulley, reaching $\frac{3}{4}$. Assuming the wrap to be $\frac{1}{2}$ the circumference, and that all the side pull of the belt comes on the rim, none being transmitted through the arms to the hub, we then have one-half of the rim pressed hard against the other half by a force equal to the resultant of the belt tensions, which, in this case, would be the sum of them. Dividing the pulley by a plane through its center and perpendicular to the belt, the cross-section of the rim cut by this plane has to take this compressive stress.

This analysis is satisfactory from an ideal standpoint only, for the intensity of stress due to the direct pull of the belt, with the usual practical proportions of rim, would be very small. Moreover, the element of speed has not been considered.

When the pulley is under speed, a set of conditions which complicates matters is introduced. The centrifugal force due to the weight of the rim and arms is no longer negligible, but has an important influence upon the design and material used. This centrifugal force acts against the effect of the belt wrap, tending to reduce the compressive stress, or, overcoming the latter entirely, sets up a tensional stress both in the rim and in the arms. It also tends to distort the rim from a true circle by bowing out the rim between the arms, thus producing a bending moment in the rim, maximum at the points where the rim joins each arm.

It can readily be imagined that the analysis in detail of these various stresses in the rim acting in conjunction with each other is quite complicated — far too much so in fact, to be introduced here. As in most cases of such design, however, one controlling influence can be separated out from the others, and the design based thereon with sufficient margin of strength to satisfy the more obscure conditions. This is rational treatment, and the "theory" will be studied accordingly.

The rim, being fastened to the ends of the arms, tends, when driving, to be sheared off, the resisting area being the areas of the cross-sections of the arms at their point of joining the rim. The force that produces this shearing tendency is the driving force of the belt, or the difference between the tensions of the tight and loose sides.

Again, at the point of connection of the arms to the hub, a shearing action takes place, so that, if this shearing tendency were carried to rupture, the hub would literally be torn out of the arms. Now, viewing the arms as beams loaded at the end with the driving force of the belt, and fixed at the hub, a heavy bending stress is set up, which is maximum at the point of connection to the hub. If the rim were stiff enough to distribute this driving force equally between the arms, each arm would take its proportional share of the load. The rim, however, is quite thin and flexible; and it is not safe to assume this perfect distribution. It is usual to consider that one-half the whole number of arms take the full driving force.

THEORY—Pulley Rim. Evidently it is practically impossible to make so thin a rim that it will collapse under the pull of a belt. As far as the *theory* of the rim is concerned, its proportion probably depends more upon the calculation for centrifugal force than upon anything else.

In order to separate this action from that of any other forces, let us suppose that the rim is entirely free from the arms and hub, and is rotating about its center. Every particle, by centrifugal force, tends to fly radially outward from the center. This condition is represented in Fig. 19. The tendency with which one-half of the rim tends to fly apart from the other is indicated by the force C_1; and the relation between C_1 and the small radial force c for each unit-length of rim can readily be found from the principles of mechanics. The case is exactly like that of a boiler or a thin pipe subjected to uniform internal pressure, which, if carried to rupture, would split the rim along a longitudinal seam.

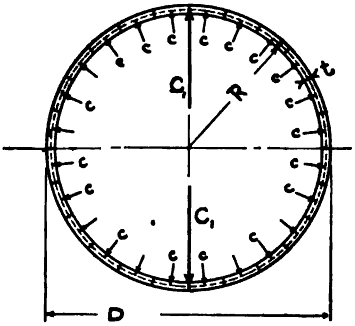


Fig. 19.

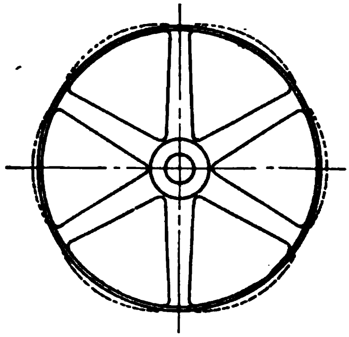


Fig. 20.

The tensile stress thus induced per square inch can be found by simple mechanics to be:

$$p = \frac{12wv^2}{g}; \qquad (17)$$

or, since $w = 0.26$ pound, and $g = 32.2$ feet per second,

$$p = 0.097\,v^2 \quad (\text{say } \frac{v^2}{10}); \qquad (18)$$

and, if p be taken equal to 1,000 pounds per square inch, which is as high as it is safe to work cast iron in this place,

$$v = 100 \text{ feet per second.} \qquad (19)$$

This shows the curious fact that the intensity of stress in the rim is directly proportional to the square of the linear velocity, and wholly independent of the area of cross-section. It is also to be noted that 100 feet per second is about the limit of speed for cast-iron pulleys to be safe against bursting.

If we wish to consider theoretically the rim together with the arms as actually connected to it, we get a much more complicated relation. This condition is shown in Fig. 20, where the rim, expanding more than the arms, bulges out between them. This makes the rim act something like a continuous beam uniformly loaded; but even then the resulting stress is not clearly defined on account of the variable stretch in the arms. Investigation on this basis is not needed further than to note that it is theoretically better, in the case of a split pulley, to make the joint close to the arms, rather than in the middle of a span.

Pulley Arms. The centrifugal force developed by the rim and arms tends to pull the arms from the hub. On the belt side, this is balanced to some extent by the belt wrap, which tends to compress the arm and relieve the tension. On the side away from the belt, the centrifugal action has full play, but the arm is usually of such cross-section that the intensity of this stress is very low. It may safely be neglected.

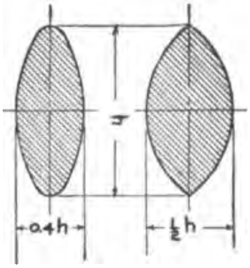


Fig. 21.

The rim being very thin in most cases, its distributing effect cannot be depended on, hence the driving force of the belt may be taken entirely by the arms immediately under the portion of the belt in contact with the pulley face. For a wrap of 180° this means that only one-half of the pulley arms can be considered as effective in transmitting the turning effort to the hub. Each of these arms is a lever fixed at one end to the hub and loaded at the other. A lever of this description is called a "cantilever" beam, its maximum moment existing at its fixed end. The load that each of these beams may be subjected to is $\frac{P}{\frac{N}{2}}$, and therefore the maximum external moment at the hub is $\frac{2PL}{N}$. From mechanics we know that the internal moment of resistance of any beam section is $\frac{SI}{c}$, and that equilibrium of the beam can be satisfied only when the external moment is equal to the internal moment of resistance of the beam section. Equating these two, we have:

$$\frac{2PL}{N} = \frac{SI}{c}. \tag{20}$$

The arms of a pulley are usually of the elliptical or segmental cross-section, and may be of the proportions shown in Fig. 21. For either of these sections the fraction $\frac{I}{c}$ is approximately equal to $0.0393h^3$. For convenience (the error caused being on the safe side), L may be taken as equal to the full radius of the pulley R, whence

$$\frac{2PR}{N} = \frac{2(T_n - T_s)R}{N} = 0.0393Sh^3, \tag{21}$$

in which S may be from 2,000 to 2,250 for cast iron

Taking moments about the center of the pulley, and solving for P_2, the force acting at the circumference of the hub, we have:

$$\frac{2PR}{N} = \frac{P_2D_1}{2};$$

or

$$P_2 = \frac{4PR}{ND_1} \tag{22}$$

The area of an elliptical section is π times the product of the half axes. With the proportions of Fig. 21, this becomes:

$$a = \pi \times 0.2h \times 0.5h = \frac{\pi h^2}{10} \tag{23}$$

Equating the external force to the internal shearing resistance, we have:

$$\frac{4PR}{ND_1} = \frac{\pi h^2 S_s}{10}$$

or,

$$S_s = \frac{40PR}{D_1N\pi h^2}. \tag{24}$$

in which the shearing stress S_s may run from 1,500 to 1,800 for cast iron.

Although both bending and shearing stresses as calculated above exist at the base of the arms, the bending is, in practically every case, the controlling factor in the design of the arms. An arm-section large enough to resist bending would have a very low intensity of shear.

If the number of arms be increased indefinitely, we come to a continuous arm or web, in which the bending action is eliminated. It may still shear off at the hub, where the area of metal is the least, at minimum circumference. In this case the area under shearing stress is $\pi D_1 T$; and the force at the circumference of the hub, is

$$\frac{PR}{\frac{D_1}{2}} = \frac{2PR}{D_1}$$

Equating external force to internal shearing resistance, we have:

$$\frac{2PR}{D_1} = \pi D_1 T S_s;$$

$$\text{or,} \quad S_s = \frac{2PR}{\pi D_1^2 T}. \qquad (25)$$

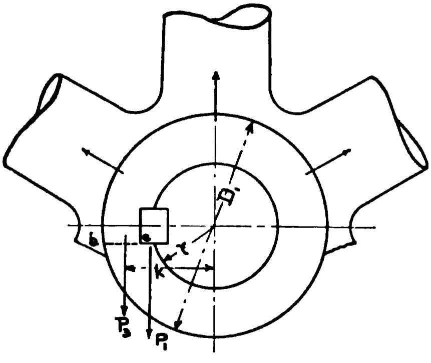


Fig. 22.

Pulley Hub. As in the case of the arms, centrifugal force does not play much part in the design of the hub of a pulley. The hub is designed principally to carry the key, and through it transmit the turning moment to the shaft. Considered thus, the hub may tear along the line of the key or crush in front of the key.

For example, in Fig. 22, if the connection with the lower arms be neglected, and the upper arms be held fast while a turning force P_1, at the surface of the shaft, is transmitted to the hub through the key, then the metal of the hub directly in front of the key is under crushing stress; and the metal along the line *eb*, from the corner to the outside, is under tensile stress. This condition is the worst that could possibly happen, because the bracing effect of the lower arms has been neglected, and the key is located between the arms.

Taking moments about the center of the shaft, the value of the force at the shaft circumference, or the "key pull," is:

$$P_1 = \frac{P R.}{r} \qquad (26)$$

Now $\frac{P_1}{P_3} = \frac{k}{r}$, k being the distance from the center of shaft to center of eb, and the area of metal which is subjected to the tearing action P_3 is $l \times eb$. Equating the external force to the internal resistance, and assuming that the stress is equally distributed over the area $l \times eb$, we have:

$$P_3 = \frac{r}{k} P_1 = \frac{r}{k} \times \frac{P R}{r} = S \times l \times e b;$$

or,

$$S = \frac{PR}{k \times l \times eb}. \qquad (27)$$

The intensity of crushing on the metal in front of the key, due to force P_1, depends upon the thickness of the key, and is properly discussed later under "Keys."

PRACTICAL MODIFICATION—Pulley Rim. The theoretical calculation for the thickness of the rim may give a thickness that could not be cast in the foundry, and the section in that case will have to be increased. As light a section as can be readily cast will usually be found abundantly strong for the forces it has to resist. A minimum thickness at the edge of the rim is about $\frac{3}{16}$ inch; and as the pulleys increase in size, the rim also must be made thicker; otherwise the rim will cool so much more quickly than the arms, that the latter, on cooling, will develop shrinkage cracks at the point of junction.

For a velocity of 6,000 feet per minute, we find from equation 18 that the tension in pounds per square inch, in the rim, due to centrifugal force, is 970. Though this in itself is a low value, yet the uncertain nature of cast iron, its condition of internal stress, due to casting, and the likely existence of hidden flaws and pockets, have established the usage of this figure as the highest safe limit for the peripheral speed of cast-iron pulleys. It is easily remembered that *cast-iron pulleys are safe for a linear velocity of about one mile per minute.*

To prevent the belt from running off the pulley, a "crown" or rounding surface is given the rim. A tapered face, which is more easily produced in the ordinary shop, may be used instead. This taper should be as little as possible, consistent with the belt staying on the pulley; ½ inch per foot each way from the center is not too much for faces 4 inches wide and less; while above this width ¼ inch per foot is enough. As little as ⅛ inch total crown has been found to be sufficient on a 24-inch face, but this is probably too little for general service.

Instead of being "crowned," the pulley may be flanged at the edges; but flanged pulley rims chafe and wear the edge of the belt.

The inside of the rim of a cast-iron pulley should have a taper of ½ inch per foot to permit easy withdrawal from the foundry mould. This is known as "draft." If the pattern be of metal, or if the pulley be machine-moulded, the greater truth of the casting does not require that the inside of the rim be turned, as the pulley, at low speeds, will be in sufficiently good balance to run smoothly. For roughly moulded pulleys, and for use at high speeds, however, it is necessary that the rim be turned on the inside to give the pulley a running balance.

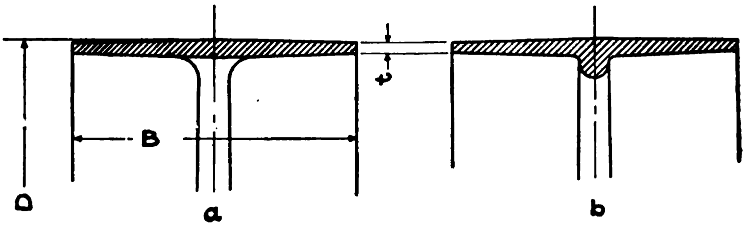


Fig. 23.

Fig. 23 shows a plain rim *a* also one stiffened by a rib *b*. Where heavy arms are used this rib is essential so that there will not be too sudden change of section at the junction of rim and arm, and consequent cracks or spongy metal.

Pulley Arms. The arms should be well filletted at both rim and hub, to render the flow of metal free and uniform in the mould. The general proportions of arms and connections to both hub and rim may perhaps be best developed by trial to scale on the drawing board. The base of the arm being determined, it may gradually taper to the rim, where it takes about the relation of $\frac{2}{3}$ to $\frac{3}{4}$ the dimensions chosen at the hub. The taper may be modified until it looks right, and then the sizes checked for strength.

Six arms are used in the great majority of pulleys. This number not only looks well, but is adapted to the standard three-jawed chucks and common clamping devices found in most shops. Elliptical arms look better than the segmental style. The flat, rectangular arm gives a very clumsy and heavy appearance, and is seldom found except on the very cheapest work.

A double set of arms may be used on an excessively wide face, but it complicates the casting to some extent.

Although a web pulley may be calculated for shear at the hub, yet it will usually be found that with a thickness of web intermediate between the thickness of the rim and that of the hub, which will satisfy the casting requirements, the requirements as to strength will be fully met.

Pulley Hub. The hub should have a taper of $\frac{1}{2}$ inch per foot draft, similar to that of the inside of the rim. The length of the hub is arbitrary, but should be ample to prevent rocking on the shaft. A common rule is to make it about $\frac{3}{4}$ the face width of the pulley.

The diameter of the hub, aside from the theoretical consideration given above, must be sufficient to take the wedging action of a taper key without splitting. This relation cannot well be calculated. Probably the best rule that exists is the familiar one that the hub should be twice the diameter of the shaft. This rule, however, cannot be literally adhered to, as it gives too small hubs for small shafts and too large ones for large shafts. It is always well to locate the key, if possible, underneath an arm instead of between the arms, thus gaining the additional strength due to the backing of the arm.

SPLIT PULLEYS.

ANALYSIS and THEORY. The split pulley is made in halves and provided with bolts through flanges and bosses on the hub for holding the two halves together. When the pulley is in place on the shaft, bolted up as one piece, it is subjected to the same forces as the simple pulley. Hence its general design follows the same principles, and we need only study the fastening of the two halves, and the effect of this fastening on the detail of rim and hub.

The simplest stress we have to consider on the rim bolts is one of pure tension, due to the centrifugal force of the halves of the pulley, A safe assumption to make is that the rim is free from the arms and hub, as in the simple pulley, and that the centrifugal force developed by it has to be taken by the rim bolts alone. In other words, consider the rim bolts as belonging entirely to the rim, and make them as strong as the rim, leaving the hub bolts to take the centrifugal force of the arms and hub, and the spreading tendency due to the key.

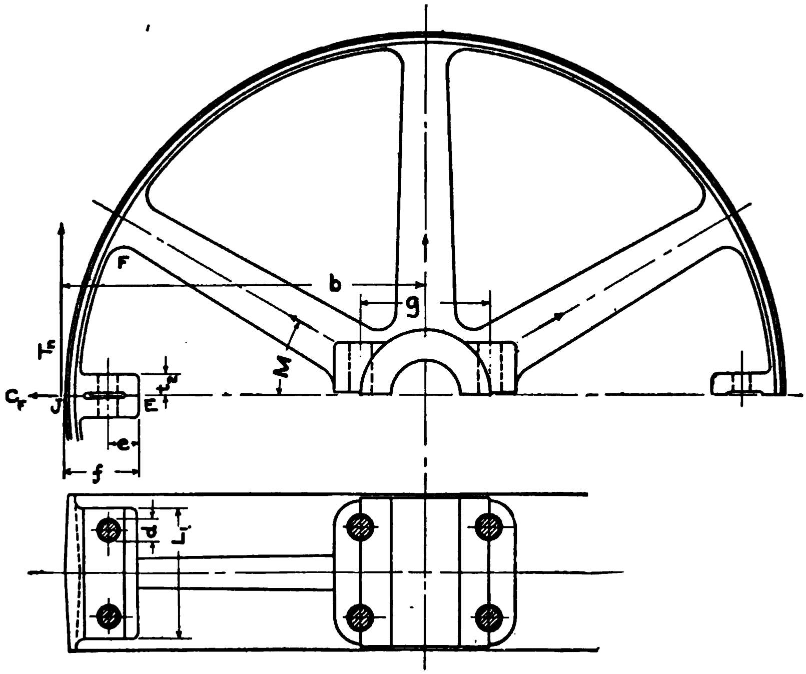


Fig. 24.

Another tensile stress is induced in the rim bolts by the fact, that, having made an open joint in the rim, and in addition placed the extra weight of lugs there, the centrifugal action at this point is increased, and at the same time a point of weakness in the rim introduced. Referring to Fig. 24, the rim flanges EJ tend to fly out due to the centrifugal force C_F. This tends to open the joint J at the outside of the rim; to throw a bending stress on the rim, maximum at the point F; and to "heel" the rim flanges about the point E. The rim bolts acting on the leverage *e* about the point E must resist these tendencies, and are thereby put in tension.

Referring to equation 18, we find the intensity of stress due to the centrifugal force of the rim in lbs. per square inch to be:

$$p = \frac{v^2}{10}.$$

If A is the sectional area of the rim in square inches, this means that the total strength of the rim is represented by $\frac{Av^2}{10}$. The strength of a bolt is represented by the expression $\frac{S\pi d_1^2}{4}$. If, now, there are n bolts in the flange, the total resisting force of the bolts is $\frac{nS\pi d_1^2}{4}$; and the equation representing equality of strength between rim and bolts is:

$$\frac{Av^2}{10} = \frac{nS\pi d_1^2}{4}, \quad (28)$$

from which, by a proper assumption of the fiber stress S, which should be low, the opening-up tendency of the joint being neglected, the diameter at the root of the thread d_1 may be calculated, and the nominal bolt diameter chosen. Reference to the table for strength of bolts, given in the chapter on Bolts, Studs, etc., will be found convenient.

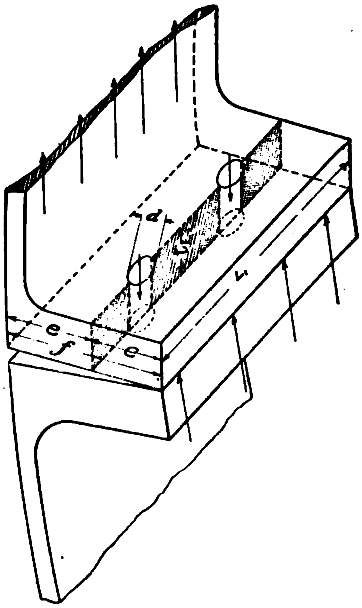


Fig. 25.

It is very doubtful if the tension on the flange bolts, due to the "heeling" about E can be calculated with sufficient accuracy to be of much value. It is probably better to assume S at a low value, say 4,000, and, in addition, for large and high-speed pulleys, stiffen the rim by running a rib between the flange and the adjacent arm. It is evident that if we make the rim so stiff that it cannot deflect, there will be no "heeling" about E; and the bolts will be well proportioned by the preceding calculation, giving them equal strength to that of the rim section.

For the bolt flange itself, any tendency to open at the joint J would cause it to act like a beam loaded at some point near its middle with the bolt load, and supported at J and E. This condition is shown in Fig. 25. Probably the weakest section would be along the line of the bolt centers. We have just noted that the carrying capacity of the bolts is $\frac{nS\pi d_1^2}{4}$. Hence, assuming that $e = \frac{1}{2}f$, which is about the worst case which could happen, we have a beam of length f loaded at the middle with $\frac{nS\pi d_1^2}{4}$ and supported at the ends. Equating the external moment to the internal moment, we have:

$$\frac{nS\pi d_1^2}{4} \times \frac{f}{4} = \frac{s(L_1 - nd)t_2^2}{6}, \qquad (29)$$

from which the fiber stress s in the flange may be calculated and judged for its allowable value.

L_1 may be assumed a little narrower than the pulley face; and t_2 from 1 inch to 2 inches or more, depending on the thickness of the rim.

The hub bolts doubtless assist the rim bolts in preventing the halves of the pulley from flying apart. They also clamp the hub tightly to the shaft, preventing any looseness on the key. Their function is a rather general one; and the specific stress which they receive is practically impossible to calculate. As a matter of fact, if the hub bolts were left out entirely, the pulley would still drive fairly well, but general rigidity and steadiness would be impaired. Hence the size of the hub bolts is more a practical question than one involving calculation. The rim bolts should be figured first, and their size determined on; then the hub bolts can be judged in proportion to the rim bolts, the diameter of shaft, the thickness and length of the hub, and the general form of the pulley. Often appearance is the deciding factor, it being manifestly inconsistent to associate small fastenings with large shafts or hubs, even though the load be actually small.

PRACTICAL MODIFICATION. Practical considerations are chiefly responsible for the location of the joint in a split pulley between the arms instead of directly at the end of an arm, where theoretically it would seem to be required. It is usually more convenient in the foundry and machine shop to have the joint between the arms; so we generally find it placed there, and strength provided to permit this. It is possible, however, to provide a double arm, or a single split arm, in which case the joint of the pulley comes at the arm, and the "heeling" action of the rim flanges is prevented.

The rim bolts should be crowded as close as possible to the rim in order to reduce the stress on them, and also the stress in the flange itself. The practical point must not be forgotten, however that the bolts must have sufficient clearance to be put into place beneath the rim.

While it is evident that the rim bolts are most effective in taking care of the centrifugal action of the halves, yet in small split pulleys it is quite common to omit the rim bolts and to use the hub bolts for the double purpose of clamping the shaft and holding the two halves together. The pulley is cast with its rim continuous throughout the full circle, and it is machined in this form. It is then cracked in two by a well-directed blow of a cold chisel, the casting being especially arranged for this along the division line by cores so set that but a narrow fin of metal holds the two parts together. This provides sufficient strength for casting and turning, but permits the cold chisel to break the connection easily.

SPECIAL FORMS OF PULLEYS.

The plain cast-iron pulley has been used in the foregoing discussion as a basis of design. A pulley is, however, such a common commercial article, and finds such universal use, that special forms, which can be bought in the open market, are not only cheaper but better than the plain cast-iron pulley, at least for regular line-shaft work.

Cast iron is a treacherous and uncertain material for rims of pulleys. It is not well suited to high fiber stresses; hence the range of speed permissible for pulley rims of cast iron is limited. Steel and wrought iron, having several times the tensional strength of cast iron, and being, moreover, much more nearly homogeneous in texture, are well suited for this work; one of the best pulleys on the market consists of a steel rim riveted to a cast-iron spider. Such an arrangement combines strength and lightness, without increasing complication or expense.

The all-steel pulley is a step further in this direction. Here the rim, arms, and hub are each pressed into shape by specially devised machinery, then riveted and bolted together. This pulley is strictly a manufactured article, which could not compete with the simpler forms unless built in large quantities, enabling automatic machinery to be used. Large numbers of pulleys are built in this way, and are put on the market at reasonable prices.

Wood-rim pulleys have been made for many years, and, except for their clumsy appearance, are excellent in many respects. The rim is built up of segments in much the same way as an ordinary pattern is made, the segments being so arranged that they will not shrink or twist out of shape from moisture. The hubs may be of cast iron, bolted to wooden webs, and carrying hardwood split bushings, which may be varied in bore within certain limits so as to fit different sizes of shafting. The wooden pulley is readily and most often used in the split form, thus enabling it to be put in position easily at any point of a crowded shaft. It is often merely clamped in place, thus avoiding the use of keys or set screws, and not burring or roughening the shaft in any way.

PROBLEMS ON PULLEYS.

1. Calculate the tensile stress due to centrifugal force in the rim of a cast-iron pulley 30 inches in diameter, at 500 revolutions per minute.

2. The driving force of a belt on a 36-inch pulley is 800 lbs., and the belt wrap about 180°. Calculate proportions of el-

liptical arms to resist bending, the allowable fiber stress being 2,000.

3. A pulley 12 inches in diameter, $\frac{5}{8}$-inch web, 4-inch diameter hub, transmits 25 horse-power at a belt speed of 3,000 ft. per minute. Calculate the maximum shearing stress in the web.

4. In Fig. 24 assume the following data: $L_1 = 7$ inches; $t_2 = 1$ inch; $e = 1\frac{1}{2}$ inches; $f' = 3$ inches; area of rim $= 3$ sq. in.; allowable tensile stress in rim 1,000 lbs. per sq. in. Calculate the diameter of the rim bolts.

5. Calculate the fiber stress in the rim bolt flange along the line of the bolts.

SHAFTS.

NOTATION—The following notation is used throughout the chapter on Shafts:

A_0 = Angular deflection (degrees).
B = Simple bending moment (inch-lbs.).
B_e = Equivalent bending moment (inch-lbs.).
c = Distance from neutral axis to outer fiber (inches).
d, d_0, d_2, d_3, d_4 = Diameters of shaft (inches).
d_1 = Internal diameter of shaft (inches).
E = Direct modulus of elasticity (a ratio).
e = Transverse deflection (inches).
G = Transverse modulus of elasticity (a ratio).
H = Horse-power (33,000 ft.-lbs. per minute).
I = Moment of inertia.
K = Distance between bearings (inches).
L = Length along shaft (inches).
L_1, L_2 = Length of bearings (inches.)
M = Distance between bearings (feet.)
N = Number of revolutions per minute.
P = Driving force of belt (lbs.).
P_1 = Load applied as stated (lbs.).
R = Radius at which load as stated acts (inches).
S = Fiber stress, tension, compression, or shearing (lbs. per sq. in.).
T = Simple twisting moment (inch-lbs.).
T_e = Equivalent twisting moment (inch-lbs.).
T_n = Tension in tight side of belt (lbs.).
T_o = Tension in loose side of belt (lbs.).
W = Load applied as stated (lbs.).

ANALYSIS. The simplest case of shaft loading is shown in Fig. 26. The equal forces W, similarly applied to the disc at the distance R from its center, tend to twist the shaft off, the tendency being equal at all points of the length L between the disc and the post, to which the shaft is rigidly fastened. The fastening to the post, of course, in this ideal case, takes the place of a resisting member of a machine. A state of pure torsion is induced in the shaft; and any element, such as *ca*, is distorted to the position *cb*, *aob* being the angular deflection for the distance L.

The case of Fig. 27 is illustrative of what occurs when a belt pulley is substituted for the simple disc. Here the twisting action is caused by the driving force of the belt, which is $T_n - T_o = P$, acting at the radius R. Torsion and angular deflection exist in the shaft, as in Fig. 26. In addition, however, another stress of a different kind has been introduced; for not only does the shaft tend to be twisted off, but the forces T_n and T_o, acting together, tend to bend the shaft, the bending moment varying with every section of the shaft, being nothing at the point *o*, and maximum at the point *c*. This combined action is the most common of any that we find in ordinary machinery, occurring in nearly every case with which we have to deal.

In Fig. 27, if the forces T_n and T_o be made equal, there will be no tendency at all to twist off the shaft, but the bending will remain, being maximum at the point *c*. This condition is illustrative of the case of all ordinary pins and studs in machines. In this sense, a pin or a stud is simply a shaft which is fixed to the frame of the machine, there being no tendency to turning of the pin or stud itself. The same condition would be realized if the disc in Fig. 27 were loose upon the shaft. In that case, the bending moment would be caused by $T_n + T_o$ acting with the leverage L. Of course there would have to be some resistance for $T_n - T_o$ to work against, in order that torsion should not be transmitted through the shaft. This condition might be introduced by having a similar disc lock with the first one by means of lugs on its face, thus receiving and transmitting the torsion.

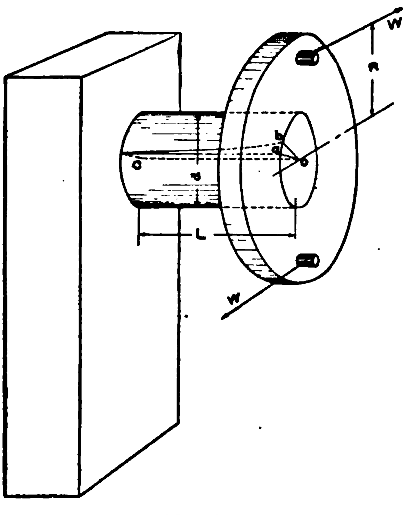


Fig. 26.

If the distance L becomes very great, both the angular deflection due to twisting, and the sidewise deflection due to bending, become excessive, and not permissible in good design. This trouble is remedied by placing a bearing at some point closer to the disc, which, as it decreases L, of course, decreases the bending moment and therefore the transverse deflection. The angular deflection can be decreased only by bringing the resistance and load nearer together.

The above implies, of course, that the diameter of the shaft is not changed, it being obvious that increase of diameter means increase of strength and corresponding decrease of both angular and transverse deflection.

If the speed of the shaft be very high, and the distance between bearings, represented by L, be very great, the shaft will take a shape like a bow string when it is vibrated, and smooth action cannot be maintained.

It is necessary to carry the cases of Figs. 26 and 27 but a single step farther to illustrate the actual working conditions of shafting in machines. Suppose the rigid post to have the shaft passing clear through it, and to act as a bearing, so that the shaft can freely rotate in it, the resistance being exerted somewhere beyond. The twisting moment will be unchanged, also the bending moment; but the effect of the bending moment will be on each particle of the shaft in succession, now putting compression on a given particle, and then tension, then compression again, and so on, a complete cycle being performed for each revolution. This

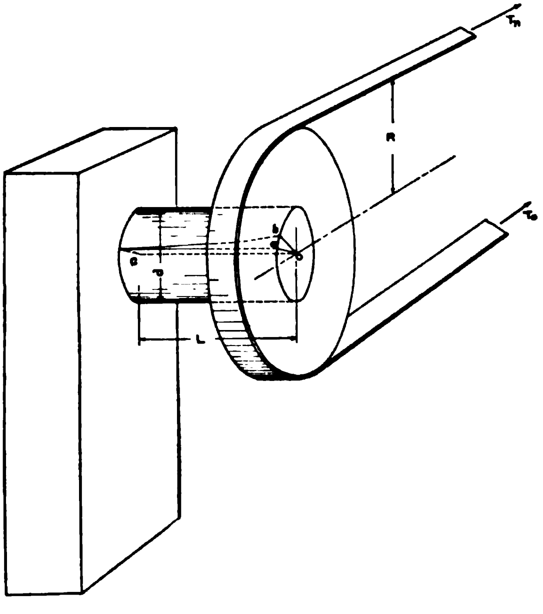


Fig. 27.

brings out a very important difference between the bending stress in pins and the bending stress in rotating shafts. In the one case the bending stress is non-reversing; in the other, reversing; and a much higher fiber stress is permissible in the former than in the latter.

THEORY—Simple Torsion. In the case of simple torsion the stress induced in the shaft is a shearing one. The external moment acts about the axis of the shaft, or is a polar moment; hence in the expression for the moment of the internal forces, the polar moment of inertia must be used. Now, from mechanics we have:

$$T = \frac{SI}{c};$$

and $$\frac{I}{c} = \frac{d^3}{5.1} \text{ (for circular section of diameter } d\text{);}$$

therefore, $$T = \frac{Sd^3}{5.1}, \qquad (30)$$

from which the diameter for any given twisting moment and fiber stress can readily be found.

For a hollow shaft this expression becomes:

$$T = \frac{S(d_o^4 - d_1^4)}{5.1d_o}. \qquad (31)$$

Simple Bending. The stresses induced in a pin or shaft under simple bending are compression and tension. The external moment in this case is transverse, or about an axis across the shaft; hence the direct moment of inertia is applicable to the equation of forces.

$$B = \frac{SI}{c};$$

and $$\frac{I}{c} = \frac{d^3}{10.2} \text{ (for circular section of diameter } d\text{);}$$

therefore, $$B = \frac{Sd^3}{10.2}. \qquad (32)$$

For a hollow shaft or pin this expression becomes:

$$B = \frac{S(d_o^4 - d_1^4)}{10.2d_o}. \qquad (33)$$

Combined Stresses. In the greater number of cases met with in practice, we find two or more simple stresses acting at the same time, and, although the shaft may be strong enough for any one of them alone, it may fail under their combined action. The most common cases are discussed below.

Tension or Pressure Combined with Bending. In Fig. 28, the load W produces a tension acting over the whole area of d, due to its direct pull. It also produces a bending action due to the leverage R, which puts the fibers at B in tension and those at the opposite side in compression. It is evident, therefore, that by taking the algebraic sum of the stresses at either side we shall obtain the net stress. It is also evident that the greatest and controlling stress will occur on the side where the stresses add, or on the tension side. Hence, from mechanics,

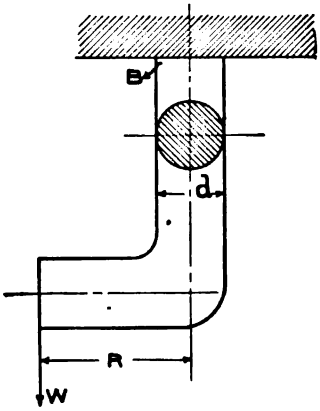


Fig. 28.

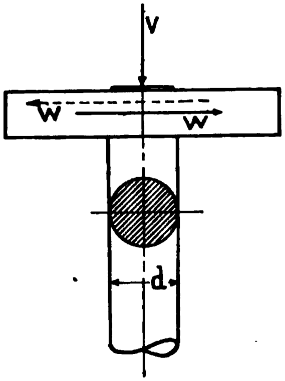


Fig. 29

$$W = \frac{\pi d^2 S}{4};$$

or,
$$S = \frac{4W}{\pi d^2} \text{ (due to direct tension).} \quad (34)$$

Also,
$$WR = \frac{S d^3}{10.2};$$

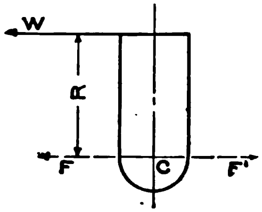
or,
$$S = \frac{10.2\ WR}{d^3} \text{ (due to bending).} \quad (35)$$

Hence the combined tensional stress acting at the point B, or, in fact, at any point on the extreme outside of the vertical shaft toward the force W, is:

$$S = \frac{4W}{\pi d^2} + \frac{10.2\ WR}{d^3}. \qquad (36)$$

If W acted in the opposite direction, the greatest stress would still be at the side B, but would be a compression instead of a tension, of the same magnitude as before.

Tension or Compression Combined with Torsion. In Fig. 29, V might be the end load on a vertical shaft; and the two forces W might act in conjunction with it as in the case of Fig. 26, at the radius R. This case is not very often met with. It is usually possible to combine the moments, find an equivalent moment of a simple kind, and use the corresponding simple fiber stress. In the case in question we have a direct stress to be combined with a shearing stress, and mechanics gives us the following solution:



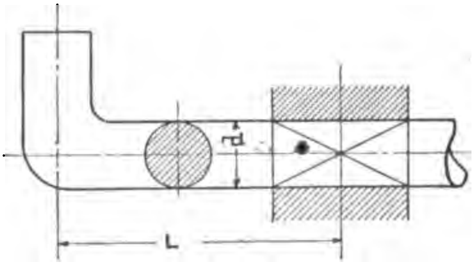


Fig. 30.

Let S_s = simple shearing stress (lbs. per sq in.).
Let S_c = simple compressive stress (lbs. per sq. in.).
Let S_{rs} = resultant shearing stress (lbs. per sq. in.).
Let S_{rc} = resultant compressive stress (lbs. per sq. in.).

We then have :

$$2WR = \frac{S_s d^3}{5.1};$$

or,

$$S_s = \frac{5.1(2WR)}{d^3}. \qquad (37)$$

Also,

$$V = \frac{\pi d^2 S_c}{4};$$

or,
$$S_c = \frac{4}{\pi d^2} V. \qquad (38)$$

Now, from a solution given in simplest form in "Merriman's Mechanics"—which the student may consult, if desired—values for the resultant stresses can be found. Whichever of these is the critical one for the material used, should form the basis for its diameter:

$$S_{rs} = \sqrt{S_s^2 + \frac{S_c^2}{4}}. \qquad (39)$$

Also,
$$S_{rc} = \frac{S_c}{2} + \sqrt{S_s^2 + \frac{S_c^2}{4}}. \qquad (40)$$

Bending Combined with Torsion. In Fig. 30, the load W acts not only to twist the shaft off, but also presses it sidewise against the bearing. As it is usually customary to figure the maximum moment as taking place at the center of the bearing, the length L, which determines the bending moment, is taken to that point. The theory of the stress induced in this case is complicated. In order to make the magnitude of the moments clearer, let us introduce the two equal and opposite forces F and F^1, each equal to W, at the point C. We can evidently do this without changing the equilibrium of the shaft in any way. We now see that W and F^1 act as a couple giving a twisting moment WR; and that F acts with a leverage L, producing a bending moment FL = WL, at the middle of the bearing.

If, now, we find an equivalent twisting moment, or an equivalent bending moment, which would produce the same effect on the fibers of the shaft as the two combined, we can treat the calculation of the diameter as a simple case, and proceed as in the cases of simple torsion and simple bending considered above. This relation is given us in mechanics:

$$B_e = \frac{B}{2} + \frac{1}{2}\sqrt{B^2 + T^2}. \qquad (41)$$

$$T_e = B + \sqrt{B^2 + T^2}. \qquad (42)$$

These expressions are true in relation to each other, on the assumption that the allowable fiber stress S is the same for tension, compression, and shearing. For the material of which shafts are usually made, this is near enough to the truth to give safe and practical results. Using the expressions for internal moments of resistance as previously noted for circular sections, we then have :

$$B_e = \frac{Sd^3}{10.2}. \qquad (43)$$

Also,

$$T_e = \frac{Sd^3}{5.1}. \qquad (44)$$

Either equation may be used ; the diameter d will result the same whichever equation is taken. For the sake of simplicity, equation 42 is generally preferred, equation 44 being taken in conjunction with it.

The expression $\sqrt{B^2 + T^2}$ is one that would be a long and tedious task to calculate. By inspection it is readily seen that this quantity can be graphically represented by means of a right-angled triangle having B and T as the sides. We may then lay down on a piece of paper, to some convenient scale, the moments B and T as the sides of a right-angled triangle, when, upon measuring the hypothenuse, we can easily read off to the same scale $\sqrt{B^2 + T^2}$. Even if the drawing is made to a small scale, the accuracy of the reading will be sufficient to enable the value for d to be solved very closely. This graphical method is illustrated in Part I.

Deflection. For a shaft subjected to pure torsion, as in Fig. 26, the angular deflection due to the load may be carried to a certain point before the limit of working fiber stress is exceeded. The equation worked out from mechanics for this condition, is:

$$A^\circ = \frac{584\ TL}{Gd^4}, \qquad (45)$$

which at once gives the number of degrees of angular deflection for a shaft whose modulus of elasticity, torsional moment, and length are known.

The shearing modulus of elasticity of ordinary shaft steel runs from 10,000,000 to 13,000,000, giving as an average about 12,000,000.

By the well-known relation of "Hooke's law" (stresses proportional to strains within the elastic limit of the material), we have:

$$\frac{A^\circ}{360} = \frac{SL}{\pi G d};$$

or

$$S = \frac{A\pi G d}{360\ L}. \qquad (46)$$

A twist of one degree in a length of twenty diameters is a usual allowance. Substituting $A = 1$, $L = 20d$, and $G = 12,000,000$, we have:

$$S = 5,240 \text{ (nearly)}. \qquad (47)$$

This is a safe value for shearing fiber stress in steel. In fact, in calculations for strength, even for reversing stresses, the usual figure is 8,000 (lbs. per square inch), thus indicating that the relation of one degree to twenty diameters is well within the limit of strength.

For a hollow shaft the above formula becomes :

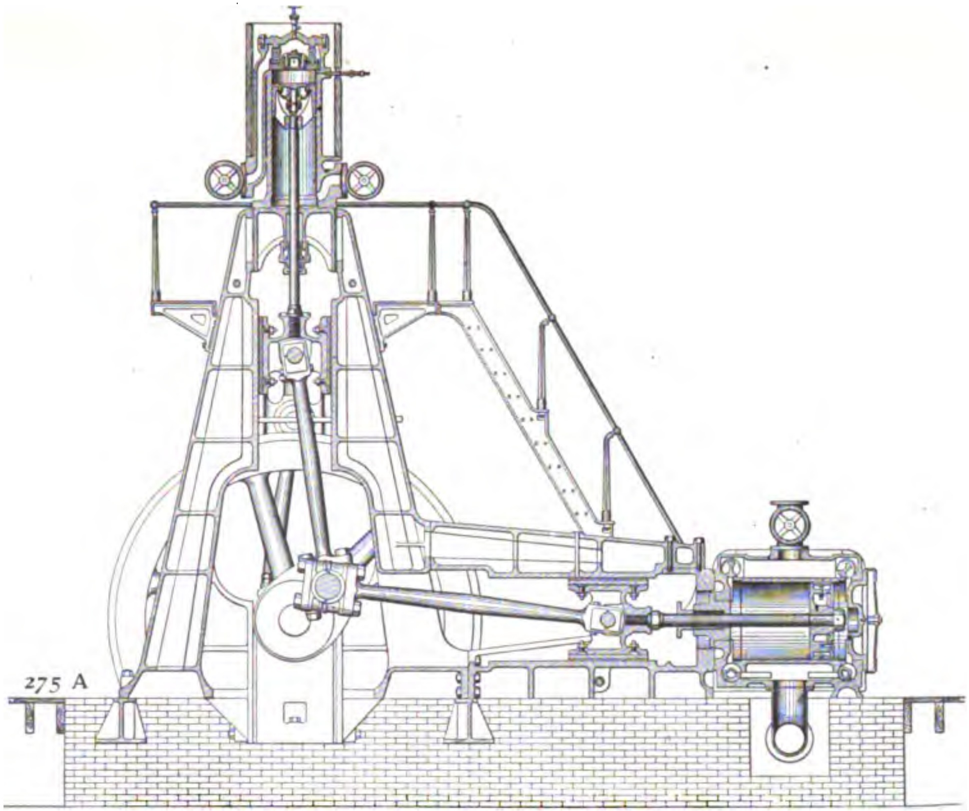
$$A^\circ = \frac{584\ TL}{G(d_0^4 - d_1^4)}. \qquad (48)$$

Transverse deflection occurs when the shaft is subjected to a bending moment. It may therefore exist alone or in conjunction with angular deflection. Transverse deflection of shafts, however, rarely exists up to the point of limiting fiber stress, because before that point is reached the alignment of the shaft is so disturbed that it is not practicable as a device for transmitting power. A transverse deflection of .01 inch per foot of length is a common allowance ; but it is impossible to fix any general limit, as in many cases this figure, if exceeded, would do no harm, while in others—such as heavily loaded or high-speed bearings—even the figure given might be fatal to good operation.

The formula for transverse deflection, deduced from mechanics, varies with the system of loading. The three most common conditions only are given below, reference to the handbook being necessary if other conditions must be satisfied:

Fixed at one end, loaded at the other,

$$e = \frac{WL^3}{3\ EI}. \qquad (49)$$



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DIRECT DRIVEN VERTICAL AMMONIA COMPRESSOR SINGLE ACTING.

Supported at ends, loaded in middle,

$$e = \frac{WL^3}{48EI}. \qquad (50)$$

Supported at ends, loaded uniformly,

$$e = \frac{5WL^3}{384EI}. \qquad (51)$$

For transverse deflection the direct modulus of elasticity must be used, for the fibers are stretched or compressed, instead of being subjected to a shearing action. The most usual value of the direct modulus of elasticity for ordinary steel is 30,000,000, and is denoted in most books by the symbol E. Both the shearing and direct moduli of elasticity are really nothing but the ratio of the stress to the strain produced by that stress, it being assumed that the given material is perfectly elastic. A material is supposed to be perfectly elastic up to a certain limit of stress, and it is within this limit that the relation as above holds good.

Expressed in the form of an equation this would be :

$$E = \frac{S}{\frac{e}{L}} = \frac{SL}{e}. \qquad (52)$$

Centrifugal Whirling. If a line shaft deflect but slightly, due to its own weight, or the weight or pressure of other bodies upon it, and then be run at a high speed, the centrifugal force set up increases the deflection, and the shaft whirls about the geometrical line through the centers of the bearings, causing vibration and wear in the adjoining members. It is evident that the practical remedy for this tendency in a shaft of given diameter and speed is to locate the bearings sufficiently close to render the action of small effect.

Many formulæ might be given for this relation, each being based on different assumptions. Perhaps as widely applied and as simple as any, is the "Rankine" formula, which sets the limit of length between bearings for shafts not greatly loaded by intermediate pulleys or side strains :

$$M = 175 \sqrt{\frac{d}{N}}. \qquad (53)$$

Horse-Power of Shafting. Horse-power is a certain specific rate of doing work, *viz.*, 33,000 foot-pounds per minute. Hence, to find the horse-power that a shaft will transmit, we must first find the work done, and then relate it to the speed. Take, for example, the case of a pulley, the symbols being the same as before—namely, P = driving force at rim of pulley (lbs.); R = radius of pulley (inches); N = number of revolutions per minute; and H = horse-power. Then,

$$\text{Work} = \text{force} \times \text{distance} = P \times (2\pi RN) = H \times 33{,}000 \times 12;$$

or,
$$PR = \frac{63{,}025H}{N}. \qquad \textbf{(54)}$$

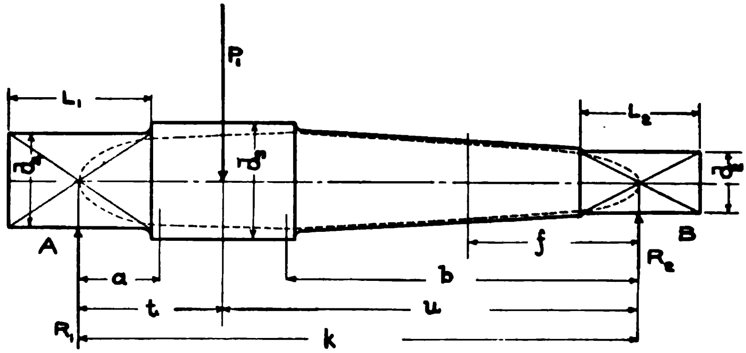


Fig. 31.

This is one of the most useful equations for calculations involving horse-power. By it the number of inch-pounds torsion for any horse-power can be at once ascertained.

It should be clearly noted, however, that in this equation the bending moment does not enter at all. Hence any shaft based in size on *horse-power alone*, is based on *torsional moment alone*, bending moment being entirely neglected. In many cases the bending moment is the controlling one as to limiting fiber stress. Hence empirical shafting formulæ depending upon the horse-power relation are unsafe, unless it is definitely known just what torsional and bending moments have been assumed.

The only safe way to figure the size of a shaft is to find accurately what torsional moment and bending moment it has to sustain, and then combine them according to equation 41 or 42 introducing the element of speed as basis for assumption of a high or low working fiber stress.

PRACTICAL MODIFICATION. The practical methods of handling the theoretical shaft equations have reference to the fit of the shaft within the several pieces upon it. The running fit of a shaft in a bearing is usually considered to be so loose that the shaft could freely deflect to the center of the bearing. This is doubtless an extreme view of the case, but it is the only safe assumption. Hence a shaft running in bearings (see Fig. 31) is supposed to be supported at the centers of those bearings, and its theoretical strength is based on this supposition.

For a tight or driving fit upon the shaft, a safe assumption to make is that there is looseness enough at the ends of the fit to permit the shaft to be stressed by the load a short distance within the faces of the hub, say from ½ inch to 1 inch. For example, referring to Fig. 31, suppose P_1 to be the transverse load, exerted through a hub fast upon the part of the shaft d_3. Taking moments about the center of one bearing, and solving for the reaction at the center of the other, we have:

$$P_1 u = R_1 K;$$

or,

$$R_1 = \frac{P_1 u}{K}. \tag{55}$$

Also,

$$P_1 t = R_2 K;$$

or,

$$R_2 = \frac{P_1 t}{K}. \tag{56}$$

Now, as far as the part of shaft d_3 is concerned, it may depend for its size on the bending moment $R_2 b$, or on $R_1 a$. The reason the lever arm is not taken to the point directly under the load P_1, is because it is not practically possible to break the shaft at that point, on account of the reinforcement of the hub, which is tightly fitted upon it. Trying these moments to see which is the greater, we shall find that the greater moment always occurs in connection with the longer lever arm. Hence $R_2 b$ will be greater than $R_1 a$. We then write the equation of *external moment = internal moment:*

$$R_2 b = \frac{S d_3^3}{10.2};$$

or,
$$d_3 = \sqrt[3]{\frac{10.2\ R_2 b}{S}}. \qquad (57)$$

For the size of bearing A we have the maximum bending moment:

$$R_1 \frac{L_1}{2} = \frac{S\, d_4^3}{10.2};$$

or,
$$d_4 = \sqrt[3]{\frac{10.2\ R_1 L_1}{2\ S}}. \qquad (58)$$

For the size of bearing B we have the maximum moment:

$$R_2 \frac{L_2}{2} = \frac{S\, d_2^3}{10.2};$$

or,
$$d_2 = \sqrt[3]{\frac{10.2\ R_2 L_2}{2\ S}}. \qquad (59)$$

The above calculations are, of course, on the assumption that no torsion is transmitted either way through this axle. We should in that case have combined torsion and bending. This has been made sufficiently clear in preceding paragraphs and in Part I, to require no further illustration.

The dotted line in Fig. 31 shows the theoretical shape the axle should take under the assumed conditions. The practical modification of this shape is obvious. At the shoulders of the shaft the corners should not be sharp, but carefully filleted, to avoid the possible starting of a crack at those points.

Often the diameter of certain parts of a shaft may be larger than strength actually calls for. For example, in Fig. 31, the part d_3 need only be as large as the dotted line; but it is obvious that unless the key is sunk in the body of the shaft, the hub could not be slipped into place over the part d_4. If, however, the diameter d_3 be made large enough so that the bottom of the key will clear d_4, the rotary cutter which forms the key way in d_3 will also clear d_4, and the key way can be more easily produced.

In cases where fits are not required to be snug, a straight shaft of cold-rolled steel is commonly used. Here any parts fastened on the middle of the shaft have to be driven over a considerable length of the shaft before they reach their final position. Moreover, there is no definite shoulder to stop against, and measurement has to be resorted to in locating them.

It does not pay to turn any portion of a cold-rolled shaft, unless it be the very ends, for relieving the "skin tension" in such material is sure to throw the shaft out of line and necessitate subsequent straightening.

Turned-steel shafts for machines may with advantage be slightly varied in diameter wherever the fit changes; and although the production of shoulders costs something, yet it assists greatly in bringing the parts to their exact location, and enables the workman to concentrate his best skill on the fine bearing fits, and to save time by rough-turning the parts that have no fits.

Hollow shafts are practicable only for large sizes. The advantages of removing the inner core of metal, aside from some specific requirement of the machine, are that it eliminates all possibility of cracks starting from the checks that may exist at the center, permits inspection of the material of a shaft, and, in case of hollow-forged shafts, gives an opening for the forging mandrel. In the last case, the material is improved by a rolling process.

The material most common for use in machine shafting is the ordinary "Machinery Steel," made by the Bessemer process. This steel is apt to be "seamy," and often contains checks and flaws that are detected only upon sudden and unexpected breakage of a part apparently sound. This characteristic is a result of the process employed in the manufacture of the steel, and thus far has never been wholly eliminated. **Bessemer steel** is, nevertheless, a very useful material, and the above weakness is not so serious but that this kind of steel can be used with success in the great majority of cases.

When a more homogeneous shaft is desired, **open-hearth steel** is available. This is a more reliable material to use than the Bessemer, and costs somewhat more. It makes a stiff, true, fine-surfaced shaft, high-grade in every respect. It is usually specified for armature shafts of dynamos and motors.

Steels of special strength, toughness, and elasticity are made under numerous processes. **Nickel steel** is perhaps the most conspicuous example. While for this steel a high price has to be paid, yet its great strength, in connection with other valuable qualities, makes it a material extremely valuable for service where light weight is essential, or where contracted space demands small size.

The range of strength of these various steels is so great that it is well-nigh useless to go into a discussion of it here. Reference should be had to the extended discussions of the handbooks, and to special trade pamphlets. A study of the possibilities of steel in its various forms for use in shafting, is very valuable as a basis for design, as it can almost be said that a machine consists chiefly of a "collection of shafts with a structure built round them." The shafts are like a core, and evidently the size of the core determines the shell about it.

PROBLEMS ON SHAFTS.

1. Required the twisting moment on a shaft that transmits 30 horse-power at 120 revolutions per minute.
2. Find the diameter of a steel shaft designed to transmit 50 horse-power at 150 revolutions per minute.
3. Assuming same data as in Problem 1, find the diameters of a hollow shaft for a value of $S = 8,000$.
4. A belt on an idler pulley embraces an angle of 120 degrees. Assuming tension of belt 1,000 pounds on each side, and pulley located midway between bearings, which are 30 inches from center to center, what is the diameter of shaft required?
5. Calculate the diameter of a steel shaft designed to transmit a twisting moment of 400,000 inch-pounds and also to take a bending moment of 300,000 inch-pounds.
6. Find the angular deflection in a 4-inch shaft 20 feet long when subjected to a load of 5,500 pounds applied to an arm of 30-inch radius. Assume transverse modulus of elasticity equal to 12,000,000.
7. The overhung crank of a steam engine has a force of 32,000 lbs. at the center of the crank pin, which is 12 inches from the center of the shaft bearing, measured parallel to the shaft. The radius of crank arm is 10 inches. Assume S equal to 10,000. Calculate the diameter of the crank shaft.
8. On a short, vertical steel shaft the load is 5,000 pounds. A gear, 36 teeth, 1½ diametral pitch, at top of shaft, transmits a load of 4,000 pounds at the pitch line. Safe shear = 7,500. What is the diameter of the shaft?

SPUR GEARS.

NOTATION—The following notation is used throughout the chapter on Spur Gears:

b = Breadth of rectangular section of arm (inches).

M, M_1 = Revolutions per minute.

μ = Coefficient of friction between teeth.

C =Width of arm extended to pitch line (inches).
c =Distance from neutral axis to outer fiber (inches).
D=Pitch diameter of gear (inches).
F=Face of gear (inches).
f =Clearance of tooth at bottom (inches).
G=Thickness of arm extended to pitch line (inches).
H=Thickness of tooth at any section (inches).
h = Depth of rectangular section of arm (inches).
I =Moment of inertia.
K=Thickness of rim (inches).
L= Distance from top of tooth to any section (inches).
N=Number of teeth.
n =Number of arms.
P=Diametral pitch (teeth per inch of diameter).
P^1=Circular pitch (inches).
Q, Q_1=Normal pressure between teeth (lbs.).
R, R_1=Resultant pressure between teeth (lbs.).
r, r_1=Radius of pitch circles (inches).
S =Fiber stress of material (lbs. per sq. in.).
s =Addendum of tooth (inches)=Dedendum of tooth.
t =Thickness of tooth at pitch line (inches).
W=Load at pitch line (lbs.).
y =Coefficient for "Lewis" formula.

ANALYSIS. If a cylinder be placed on a plane surface, with its axis parallel to the plane, an attempt to rotate the cylinder about its axis would cause it to roll on the plane.

Again, if two cylinders be provided with axial bearings, and be slightly pressed together, motion of one about its axis will cause a similar motion of the other, the two surfaces rolling one on the other at their common tangent line. If moved with care, there will be no slipping in either of the above cases—which is explained by the fact that no matter how smooth the surfaces may appear to be, there is still sufficient roughness to make the little irregularities interlock and act like minute teeth.

The magnitude of the force possible to be transmitted depends not only on the roughness of the surfaces, but on the amount of pressure between them. Suppose that one cylinder is a part of a hoisting drum, on which is wound a rope with a weight attached. We can readily make the weight so great that, no matter how hard we press the two cylinders together, the driving cylinder will not turn the hoisting cylinder, but will slip past it. If now, instead of increasing the pressure, which is detrimental both to cylinders and bearings of same, we increase the coarseness of the surfaces, or, in other words, put teeth of appreciable size on these surfaces, we attain the desired result of positively driving without excessive side pressure.

These artificial projections, or teeth, must fit into one another; hence the surfaces of the original cylinders, having been broken up into alternate projections and hollows, have entirely disappeared to the eye; they nevertheless exist as ideal or imaginary surfaces, which roll together with the same surface velocities as if in bodily form, provided that the curves of the teeth are correctly formed. Several mathematical curves are available for use as tooth outlines, but in practice the **involute** and **cycloidal curves** are the only ones used for this purpose.

The ideal surfaces are known as **pitch cylinders** or **pitch circles.** In Fig. 32 is shown an end view of such a pair of cylinders in contact at their pitch point P. In gear calculations we assume that there is no slip between the pitch circles, acting as driving cylinders; hence the speeds of the two pitch circles at the pitch point are equal. If M and M_1 be the revolutions per minute of the cylinders respectively, r and r_1 their radii, then

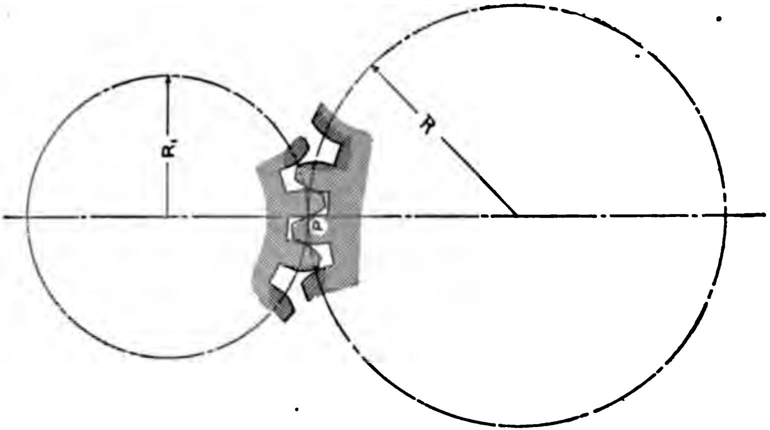


Fig. 32.

$$2 \pi r M = 2 \pi r_1 M_1;$$

or,
$$\frac{M}{M_1} = \frac{r_1}{r}. \qquad (60)$$

That is, the number of revolutions varies inversely as the radii.

The simple calculation as above is the key to all calculations involving gear trains in reference to their speed ratio.

Fig. 33 represents cycloidal teeth in the two extreme positions of beginning and ending contact. The normal pressure Q or Q_1 between the teeth in each position acts through the pitch point O, as it must always do in order to insure the condition of ideal rolling of the pitch circles, and the velocity ratio proportional to $\frac{r_1}{r}$ As the surfaces of the teeth slide together, frictional resistance is produced at their point of contact. This force is widely variable, depending on the material and condition of the tooth surfaces, whether smooth and well lubricated, or rough and gritty. As this resistance acts in conjunction with the normal force between the teeth, we may construct a parallelogram of forces on these two as a base, the resultant pressure between the teeth being slightly changed thereby, as shown in Fig 33.

Assuming a coefficient of friction μ, the force of friction is μ Q or μ Q_1 and the resultant pressure R or R_1.

Tooth B of the FOLLOWER is therefore under a heavy bending moment measured by the product RL, L being the perpendicular distance from the center of the tooth at its base to the line of the force. This tooth also has a relatively small compressive stress due to the resolved part of R along the radius, and a relatively small shearing stress due to the resolved part of R along a tangent to the pitch circle.

Tooth D of the driven wheel or FOLLOWER has a relatively large shearing stress, a small bending moment, and practically no direct compressive stress.

Tooth A of the driving wheel or DRIVER has a relatively large shearing stress, a small bending moment, and small compressive stress.

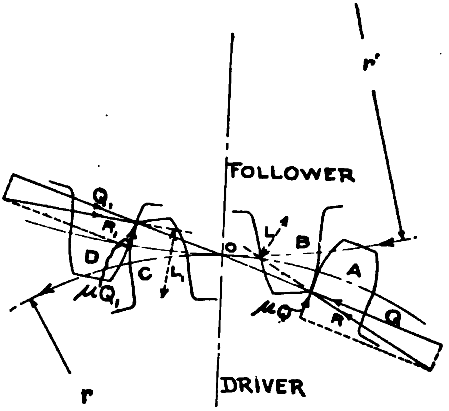


Fig. 33.

Tooth C of the DRIVER has a large bending moment, but small compressive and shearing stresses.

The conditions as noted above are not those of every pair of gears, in fact they vary with every difference of pitch circle, or of detail and position of tooth. It is true, however, that in nearly all cases in practice the bending stress is the controlling one from a theoretical standpoint. Moreover, the designer must consider the form and strength of the tooth when it is under the condition of maximum moment. This evidently, from the above, occurs at the beginning of contact, for the follower teeth; and at the end of contact, for the driver teeth. In the particular case illustrated in

Fig. 33, if the material in both gears were the same, tooth C, being the weaker at the root, would probably break before B; but if C were of steel, and B of cast iron, B might break first.

It will be noticed that R is nearly parallel to the top of the tooth; and it may easily happen that the friction may become of such a value that it will turn the direction of R until it lies along the top of the tooth exactly, which is the condition for maximum moment. For strength calculations it is usual to consider this condition as existing in all cases.

At the beginning of contact there is more or less shock when the teeth strike together, and this effect is much more evident at high speeds. There is also at the beginning of contact a sort of chattering action as the driving tooth rubs along the driven tooth.

Uniform distribution of pressure along the face of the tooth is often impaired by uneven wear of the bearings supporting the gear shafts, the pressure being localized on one corner of the tooth. The same effect is caused by the accidental presence of foreign material between the teeth. Again, in cast gearing, the spacing may be irregular, or, on account of draft on the pattern, the teeth may bear at the high points only. While it is usual to consider that the load is evenly distributed along the face of the tooth, yet the above considerations show that *an ample margin of strength must always be allowed* on account of these uncertainties.

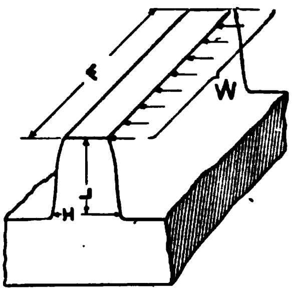


Fig. 34.

When the number of teeth in the mating gears is high, the load will be distributed between several teeth; but, as it is almost certain that at some time the proper distribution of load will not exist, and that one tooth will receive the full load, it is considered that practically the only safe method is so to design the teeth that a single tooth may be relied upon to withstand the full load without failure.

THEORY. Based on the Analysis as given, the theory of gear teeth assumes that one tooth takes the whole load, and that this load is evenly distributed along the top of the tooth and acts parallel with its base, thus reducing the condition of the tooth to that of a cantilever beam. The magnitude of this load at the top of the tooth is taken for convenience the same as the force transmitted at the pitch circle. This condition is shown in Fig. 34. Equating the external moment to the internal moment, we then have, from mechanics:

$$WL = \frac{SI}{c} = \frac{SFH^2}{6} \qquad (61)$$

The thickness H is usually taken either at the pitch line or at the root of the tooth just before the fillet begins; and L, of course, is dependent on the tooth dimensions. The formula is most readily used when the outline of the tooth is either assumed or known, a trial calculation being made to see if it will stand the load, and a series of subsequent calculations followed out in the same way until a suitable tooth is found. This method is pursued because there are certain even pitches which it is desirable to use; and it is safe to say that any calculation figured the reverse way would result in fractional pitches.. The latter course may be used, however, and the nearest even pitch chosen as the proper one.

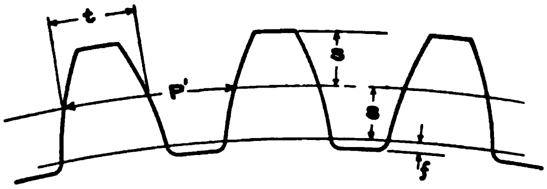
As stated under "Analysis," there are a great many circumstances attending the operation of gears which make impossible the purely theoretical application of the beam formulæ. For this reason there is no one element of machinery which depends so much on experience and judgment for correct proportion as the tooth of a gear. Hence it is true that a rational formula based on the theoretical one is really of the greater practical value in tooth design.

If we examine formula 61, we find that in a form solved for W, we have:

$$W = \frac{SFH^2}{6\,L}. \qquad (62)$$

Of these quantities, H and L are the only variables, for we can make the others what we choose. H and L depend upon the circular pitch P^1 and the curvature and outline of the tooth. If now we could settle on a standard system of teeth, we could establish a coefficient to be used to take the place of the variable part of H and L, which depends on the outline of tooth, and we should thus have an empirical formula which would be on a theoretical basis.

This, Mr. Wilfred Lewis has done; and it is safe to say that this formula is more universally used and with more satisfactory practical results than any other formula, theoretical or practical, that has ever been devised. His coefficient is known as y, and was determined from many actual drawings of different forms of teeth showing the weakest section. This coefficient is worked out for the three most common systems as follows:



$$t = \frac{P^1}{2}$$

$$f = \frac{t}{10}$$

$$s = \frac{P^1}{\pi}$$

Fig. 35.

For 20° involute, $$y = 0.154 - \frac{0.912}{N}. \qquad (63)$$

For 15° involute and cycloidal, $$y = 0.124 - \frac{0.684}{N}. \qquad (64)$$

For radial flanks, $$y = 0.075 - \frac{0.276}{N}. \qquad (65)$$

The tooth upon which the above is based is the American standard or Brown & Sharpe tooth, for which the proportions are shown in Fig. 35.

The "Lewis" formula* is:

$$W = SP^1 Fy. \qquad (66)$$

A table indicating the value of S for different speeds follows:

Safe Working Stresses for Different Speeds.

Speed of teeth, ft. per min..	100	200	300	600	900	1200	1800	2400
Cast iron	8000	6000	4800	4000	3000	2400	2000	1700
Steel	20000	15000	12000	10000	7500	6000	5000	4300

---

*Note. A full and convenient statement of the Lewis formula will be found in "Kent's Pocket Book."

A usual relation of F to P^1 is:

For cast teeth, $F = 2P^1$ to $3P^1$. **(66)**

For cut teeth, $F = 3P^1$ to $4P^1$. **(67)**

The usual method of handling these formulæ is as follows:

The pitch circles of the proposed gears are known or can be assumed; hence W can readily be figured, also the speed of the teeth, whence S can be read from the table. The desired relation of F to P^1 can be arbitrarily chosen, when P^1 and y become the only unknown quantities in the equation. A shrewd guess can be made for the number of teeth, and y calculated therefrom. Then solve the equation for P^1 which will undoubtedly be fractional. Choose the nearest even pitch, or, if it is desired to keep an even diametral pitch, the fractional pitch that will bring an even diametral pitch. Now, from this final and corrected pitch, and the diameter of the pitch circle, calculate the number of teeth N in the gear. Check the assumed value of y by this positive value of N.

Another good way of using this formula is to start with the pitch and face desired, and the diameter of the pitch circle. In this case W is the only unknown quantity, and when found can be compared with the load required to be carried. If too small, make another and successive calculations until the result approximates the required load.

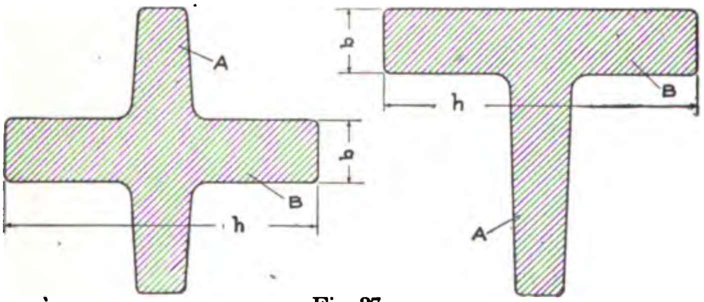


Fig. 37.

SPUR GEAR RIM, ARMS, AND HUB.

ANALYSIS and THEORY. The rim of a gear has to transmit the load on the teeth to the arms. It is thus in tension on one side of the teeth in action, and in compression on the other. The section of the rim, however, is so dependent on other practical considerations which call for an excess of strength in this respect, that it is not considered worth while to attempt a calculation on this basis.

Gears seldom run fast enough to make necessary a calculation for centrifugal force; and in general it can be said that the design of the rim is entirely dependent on practical considerations. These will appear later under "Practical Modification."

The arms of a gear are stressed the same as pulley arms, the same theory answering for both, except that a gear rim always being much heavier than a pulley rim, the distribution of load amongst the arms is better in the case of a gear than of a pulley, and it is usually safe to assume that each arm of a gear takes its full proportion of load; or, for an oval section, equating the external moment to the internal moment as in the case of pulleys, we have:

$$\frac{WD}{n2} = 0.0393\ Sh^3. \qquad (68)$$

Heavy spur gears have the arms of a cross or T section (Fig. 37), the latter being especially applicable to the case of bevel gears where there is considerable side thrust. The simplest way of treating such sections is to consider that the whole bending moment is taken by the rectangular section whose greater dimension is in the direction of the load. The rest of the section, being close to the neutral axis of the section, is of little value in resisting the direct load, its function being to give sidewise stiffness. The equation for the cross or T style of arm, then is:

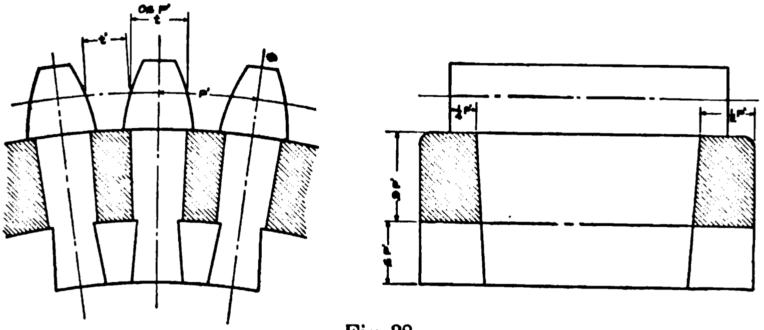


Fig. 38.

$$\frac{W}{n} \times \frac{D}{2} = \frac{Sbh^2}{6}. \qquad (69)$$

Either *b* or *h* may be assumed, and the other determined. As a guide to the section, *b* may be taken at about the thickness of the tooth.

Gear hubs are in no wise different from the hubs of pulleys or other rotating pieces. The depth necessary for providing sufficient strength over the key to avoid splitting is the guiding element, and can usually be best determined by careful judgment.

PRACTICAL MODIFICATION. The practical requirements, which no theory will satisfy, are many and varied. Sudden and severe shock, excessive wear due to an atmosphere of grit and corrosive elements, abrupt reversal of the mechanism, the throwing-in of clutches and pawls, the action of brakes—these and many other influences have an important bearing on gear design, but not one that can be calculated. The only method of procedure in such cases is to base the design on analysis and theory as previously given, and then add to the face of gear, thickness of tooth, or pitch an amount which judgment and experience dictate as sufficient.

Excessive noise and vibration are difficult to prevent at high speeds. At 1,000 feet per minute, gears are apt to run with an unpleasant amount of noise. At speeds beyond this, it is often necessary to provide **mortise teeth,** or teeth of hard wood set into a cast-iron rim (see Fig. 38). Rawhide pinions are useful in this regard. Fine pitches with a long face of tooth run much more smoothly at high speeds than a coarse pitch and narrow-faced tooth of equal strength. Greater care in alignment of shafts, however, is necessary, also stiffer supports.

Should it be impracticable to use a standard tooth of sufficient strength, there are several ways in which we can increase the carrying capacity without increasing the pitch. These are:

1. Use a stronger material, such as steel.
2. Shroud the teeth.
3. Use a hook tooth.
4. Use a stub tooth.

Shrouding a tooth consists in connecting the ends of the teeth with a rim of metal. When this rim is extended to the top of the tooth, the process is called "full-shrouding" (Fig. 39); and when carried only to the pitch line, it is termed "half-shrouding" (Fig. 40). The theoretical effect of shrouding is to make the tooth act like a short beam built in at the sides; and the tooth will practically have to be sheared out in order to fail. This modification of gear design requires the teeth to be cast, as the cutter cannot pass through the shrouding. The strength of the shrouded gear is estimated to be from 25 to 50 per cent above that of the plain-tooth type.

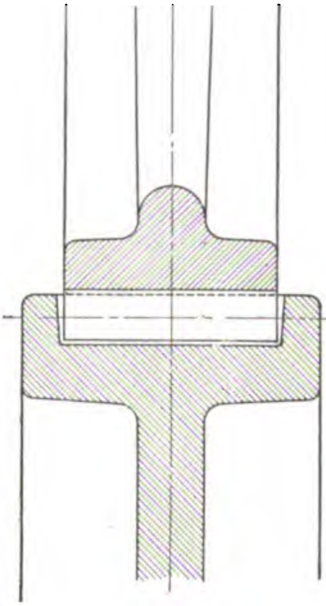


Fig. 39.

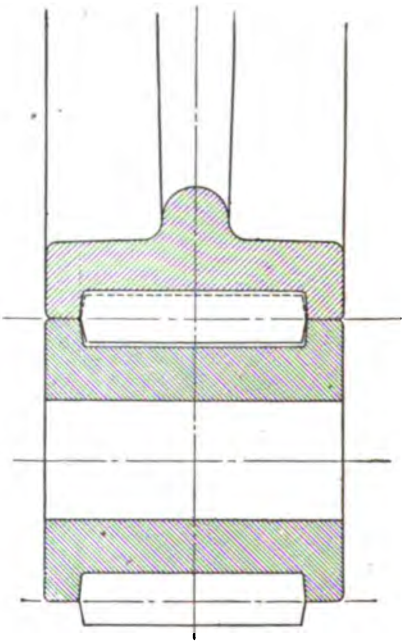


Fig. 40.

The **hook-tooth gear** (Fig. 41) is applicable only to cases where the load on the tooth does not reverse. The working side of the tooth is made of the usual standard curve, while the back is made of a curve of greater obliquity, resulting in a considerable increase of thickness at the root of the tooth. A comparison of strength between this form and the standard may be made by drawing the two teeth for a given pitch, measuring their thickness just at top of the fillet, and finding the relation of the squares of these dimensions. The truth of this relation is readily seen from an inspection of formula 61.

The **stub tooth** merely involves the shortening of the height of the tooth in order to reduce the lever arm on which the load acts, thus reducing the moment, and thereby permitting a greater load to be carried for the same stress.

The rim of a gear is dependent for its proportions chiefly on questions of practical moulding and machining. It must bear a certain relation to the teeth and arms, so that, when it is cooling in the mould, serious shrinkage stresses will not be set up, forming pockets and cracks. Moreover, when under pressure of the cutter in the producing of the teeth, it must not chatter or spring. This condition is quite well attained in ordinary gears when the thickness of the rim below the base of the tooth is made about the same as the thickness of the tooth.

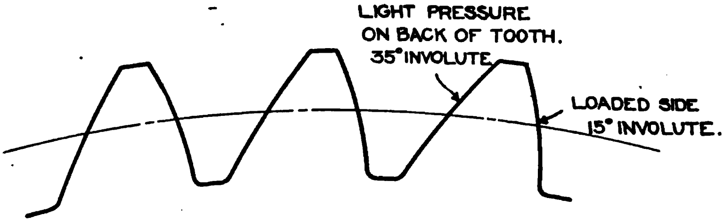


Fig. 41.

The stiffening ribs and arms must all be joined to the rim by ample fillets, and the cross-section must be as uniform as possible, to prevent unequal cooling and consequent pulling-away of the arms from the rim or hub. Often the calculated size of the arms at both rim and hub has to be modified considerably to meet this requirement.

The arms are usually tapered to suit the designer's eye, a small gear requiring more taper per foot than a large one. Both rim and hub should be tapered $\frac{1}{2}$ inch per foot to permit easy drawing-out from the mould.

The proportions given in the following table have been used with success as a basis of gear design in manufacturing practice. The table will serve as an excellent guide in laying out, and can be closely followed, in most cases with but slight modification. **Web gears** are introduced for small diameters where the arms begin to look awkward and clumsy.

Gear Design Data.

Measurements given in inches. Letters refer to Fig. 42.

Diametral pitch ..	P	1½	1¾	2	2½	3	3½	4	5	6	8
Face............	F	6¼	5½	4¾	3¾	3¼	2¾	2½	2⅛	1⅞	1½
Thickness of arm when extended to pitch line....	G	1⅜	1¼	1⅛	1	⅞	13/16	¾	11/16	⅝	½
Width of arm when extended to pitch line	C	4	3½	3	2½	2¼	2	1¾	1½	1⅜	1⅛
Thickness of rim..	K	2¾	2⅝	2⅛	1¾	1½	1⅜	1¼	1	⅞	¾
Depth of rib......	E	2	1¾	1½	1¼	1	⅞	¾	⅝	½	⅜
Thickness of web.	T	1⅛	1	⅞	¾	⅝	9/16	½	7/16	⅜	5/16

Number of arms, 6.

Give inside of rims and hub a draft of ⅛ inch per foot.

BEVEL GEARS.

NOTATION—The following notation is used throughout the chapter on Bevel Gears:

A =Apex distance at pitch element of cone (inches).

A^1=Apex distance at bottom element of tooth (inches).

B =Angle of bottom of tooth (degrees).

C =Pitch angle (degrees).

D =Pitch diameter (inches).

E =Radius increment of gear (inches).

F =Face of gear (inches).

f =Clearance at bottom (inches).

G =Angle of face (degrees).

H =Cutting angle (degrees).

K =Radius increment of pinion (inches).

N =Number of teeth.

N^1=Formative number of teeth, or the number corresponding to the spur gear on which the outline of tooth is made.

O D=Outside diameter (inches).

P =Diametral pitch related to pitch diameter (teeth per inch).

P^1 =Circular pitch measured on the circumference of D (inches).

S =Working strength of material (lbs. per sq. in.).

s =Addendum, or height of tooth above pitch line (inches).

$s+f$=Depth of tooth below pitch line (inches).

T =Angle of top of tooth (degrees).

t =Thickness of tooth at pitch line (inches).

W =Working load at pitch line (lbs.).

y =Factor in "Lewis" formula.

ANALYSIS. It is possible to consider bevel gears as the general case of which spur gears are a special form. The pitch surfaces of spur gears described above as cylinders, mathematically considered, are cones whose vertices are infinitely distant, while bevel gears likewise are based on pitch cones, but with a vertex at some finite point, common to the mating pair. Hence, as we might expect, the laws of tooth action are similar in bevel gears to those in the case of spur gears. The profile of the tooth in the former case, however, is based, not on the real radius of the pitch cone, but on the radius of the normal cone; and in the development of the outline the latter is treated just as though it were the radius of a spur gear. The tooth thus formed is wrapped back upon the normal cone face, and becomes the large end of the tapering bevel-gear tooth (see Fig. 44).

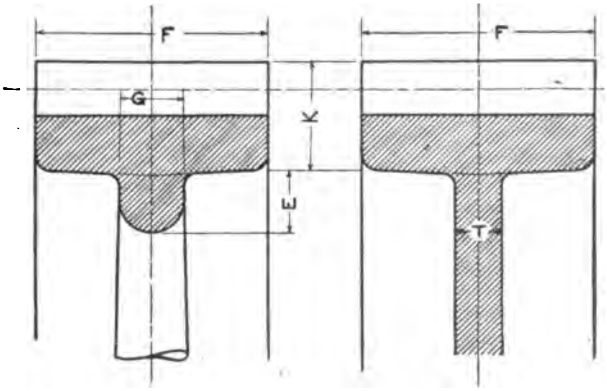


Fig. 42.

The teeth of bevel gears, being simply projections with bases on the pitch cones, have a varying cross-section decreasing toward the vertex; also a trapezoidal section of root, the latter section acting as a beam section to resist the cantilever moment due to the tooth load.

The arms must, as in the case of spur gears, transmit the load from the tooth to the shaft; in addition, the arms of a bevel gear are subjected to a side thrust due to the wedging action of the cones. Hence sidewise stiffness of the arms is more essential in this type of gear than in the case of the spur gear.

THEORY. It is evident that the calculation of tooth strength based on a trapezoidal section of root would be somewhat compli-

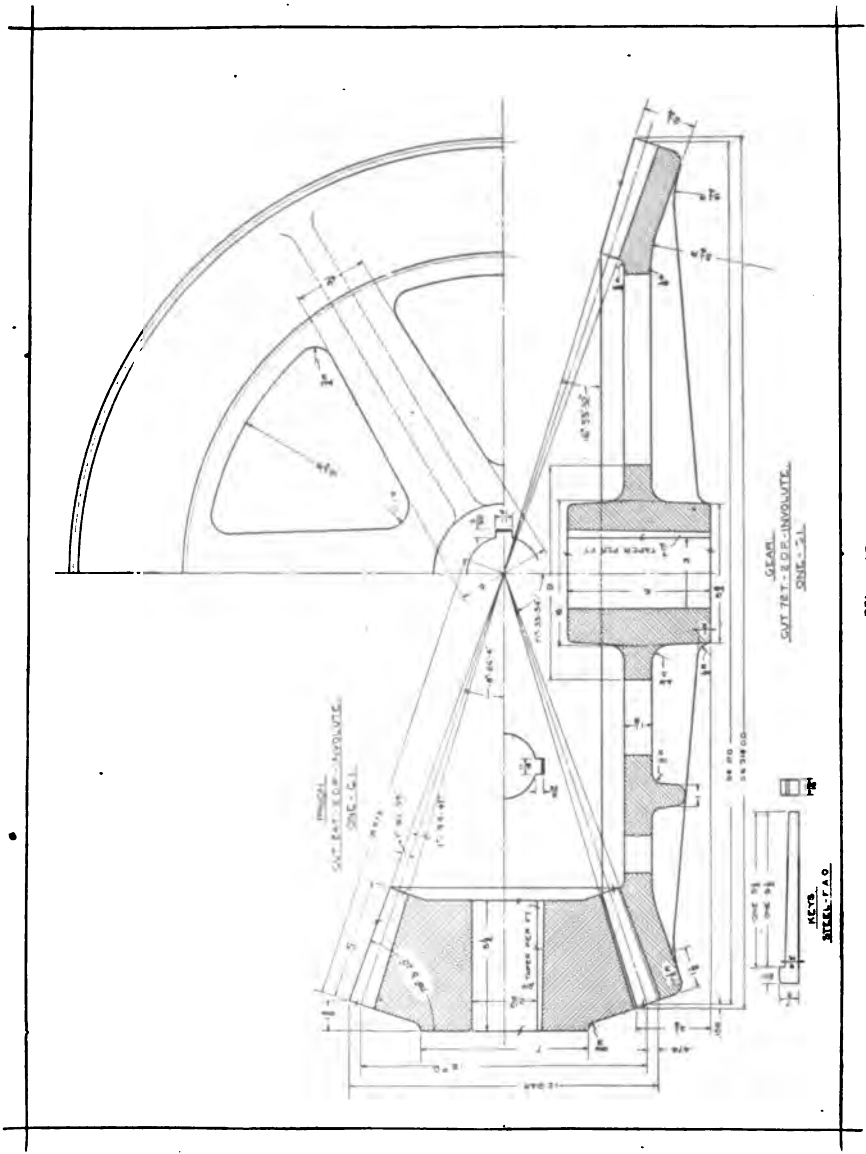

GEAR
CUT 72 T. - 2 D.P. - INVOLUTE.
ONE - C.I.
PINION
CUT 24 T. - 2 D.P. - INVOLUTE.
ONE - C.I.
KEYS
STEEL - T.A.O.

Fig. 43.

cated ; also that the trapezoid in most cases would be but little different from a true rectangle. Hence the error will be but slight if the average cross-section of the tooth be taken to represent its strength, and the calculation made accordingly.

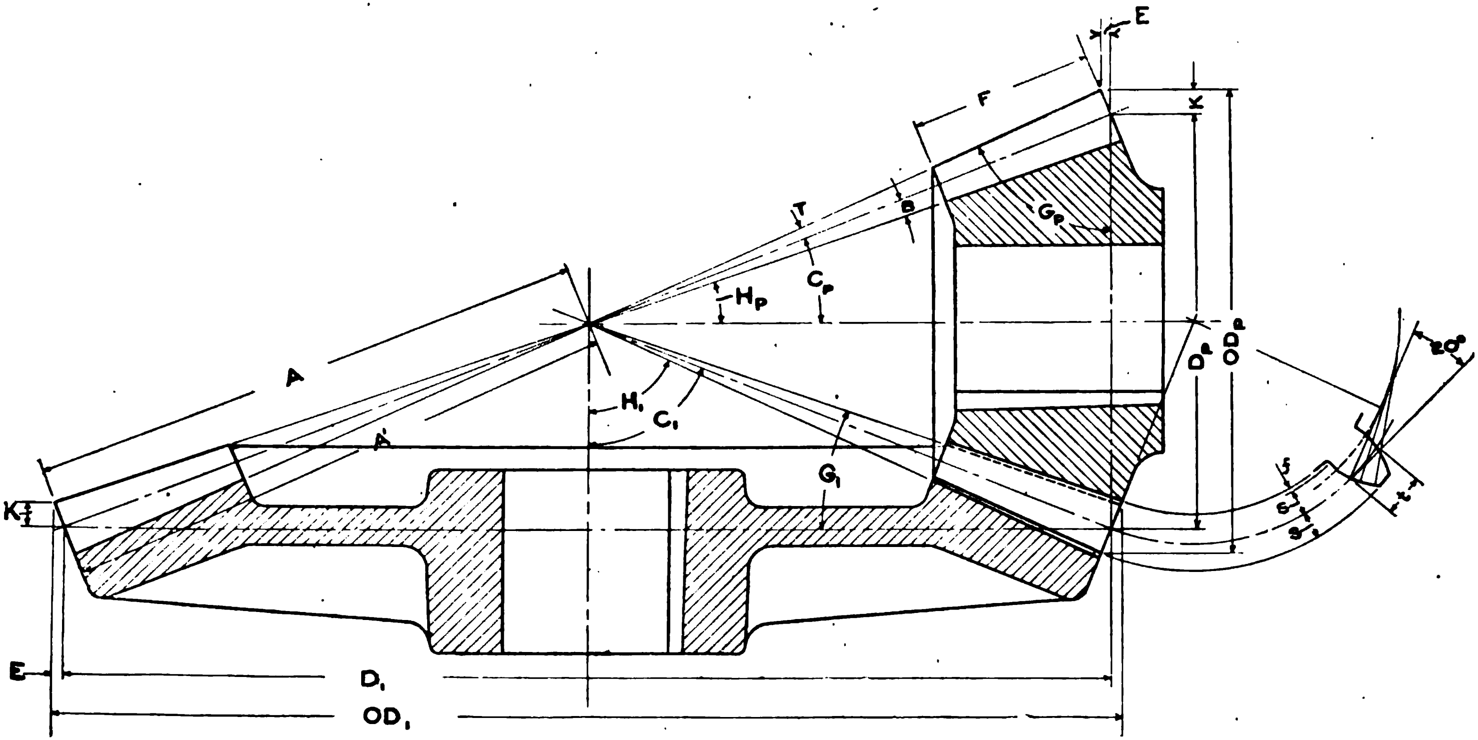


Fig. 44.

Fig. 45 shows a bevel-gear tooth with the average cross-section in dotted lines. For the purpose of calculation, the assumption is made that the section A is carried the full length of the face of the gear, and that the load which this average tooth must carry is the calculated load at the pitch line of section A. This is equivalent to saying that the strength of a bevel-gear tooth is equal to that of a spur-gear tooth which has the same face, and a section identical with that cut out by a plane at the middle of the bevel tooth. The load, as in the case of the spur gear, should be taken at the top of the tooth; and its magnitude can be conveniently calculated at the mean pitch radius of the bevel face, without appreciable error.

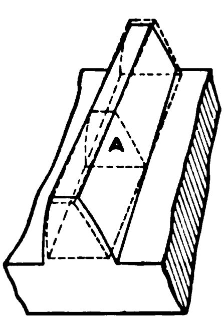


Fig. 45.

This similarity to spur gears being borne in mind, the calculation for strength needs no further treatment. Once the average tooth is assumed or found by layout, a strict following-out of the methods pursued for spur-gear teeth will bring consistent results.

The detail design of a pair of bevel gears involves some trigonometrical computations in order properly to dimension the drawing for use in finishing the blanks and subsequently in cutting the teeth, or, in the case of cast gears, in making the pattern. These calculations, although simple, are yet apt to be tedious; and inaccuracies are likely to creep in if a definite system of relations be not maintained. Hence the results of these calculations are given below in condensed and reduced form. The deduction of these formulæ is a simple and interesting exercise in trigonometry; and it is urged that they be worked out by the student from the figure, in which case he will feel greater confidence in their use.

Axes of Gears at 90 Degrees.

Use subscript 1 for gear; P for pinion. Letters refer to Fig. 44.

$$P = \frac{N}{D} = \frac{\pi}{P'}. \qquad (70)$$

$$s = \frac{1}{P} = \frac{P'}{\pi}. \qquad (71)$$

$$t = \frac{P'}{2} = \frac{\pi}{2P}. \qquad (72)$$

$$f = \frac{t}{10} = \frac{P^1}{20} = \frac{\pi}{20P}. \qquad (73)$$

$$\tan C_p = \frac{N_p}{N_1};\ \tan C_1 = \frac{N_1}{N_p}. \qquad (74)$$

$$\tan T = \frac{s}{A} = \frac{2 \sin C}{N}. \qquad (75)$$

$$\tan B = \frac{s+f}{A} = \frac{2.314 \sin C}{N}. \qquad (76)$$

$$s + f = A \tan B = \frac{1.157}{P} = 0.368P^1. \qquad (77)$$

$$A = \frac{N}{2P \sin C} = \frac{1}{2P}\sqrt{N_1^2 + N_p^2} = \frac{1}{2}\sqrt{D_1^2 + D_p^2}. \qquad (78)$$

$$A^1 = \frac{A}{\cos B} = \frac{N}{2P \cos B \sin C}. \qquad (79)$$

$$G_1 = 90^\circ - (C_1 + T);\ G_p = 90^\circ - (C_p + T). \qquad (80)$$

$$E = S \cos C_1 = S \sin C_p. \qquad (81)$$

$$K = S \cos C_p = S \sin C_1. \qquad (82)$$

PRACTICAL MODIFICATION. The practical requirements to be met in transmission of power by bevel gears are the same as for spur gears; but in the case of bevel gears even greater care is necessary to provide stiffness, strength, true alignment, and rigid supports. As far as the gears themselves are concerned, a long face is desirable; but it is much more difficult to gain the advantage of its strength than in the case of spur gears, because full bearing along the length of the tooth is hard to guarantee.

The rim usually requires a series of ribs running to the hub to give required stiffness and strength against the side thrust which is always present in a pair of bevel gears. Instead of arms, the tendency of bevel-gear design, except for very large gears, is toward a web on account of the better and more uniform connection thereby secured between rim and hub. This web may be lightened by a number of holes, so that the resultant effect is that of a number of wide and flat arms.

The hubs naturally have to be fully as long as those of spur gears, because there is greater tendency to rock on the shaft, due to the side thrust from the teeth, mentioned above.

The teeth on small gears are cut with rotary cutters, at least two finishing cuts being necessary, one for each side of the tapering tooth. The more accurate method is to plane the teeth on a special gear planer, and this method is followed on all gears of any considerable size. The practical requirement here is that no portion of the hub shall project so as to interfere with the stroke of the planer tool. The requirements of gear planers vary somewhat in this regard.

Finally, after all that is possible has been done in the design of the gear itself to render it suitable to withstand the varied stresses, especial attention must be paid to the rigidity of the supporting shafts and bearings. Bearings should always be close up to the hubs of the gears, and, if possible the bearing for both pinion and gear should be cast in the same piece. If this is not done, the tendency of the separate bearings to get out of line and destroy the full bearing of the teeth is difficult to control. Thrust washers are desirable against the hubs of both pinion and gear; also proper means of well lubricating the same.

With these considerations carefully met, bevel gears are not the bugbear of machine design that they are sometimes claimed to be. The common reason why bevel gears cut and fail to work smoothly, is that the gears and supports are not designed carefully enough in relation to each other. This is also true of spur gears, but the bevel gear will reveal imperfections in its design far the more quickly of the two.

WORM AND WORM GEAR.

NOTATION—The following notation is used throughout the chapter on Worm and Worm Gear:

D = Pitch diameter of gear (inches).
E = Efficiency between worm shaft and gear shaft (per cent).
f = Clearance of tooth at bottom (inches).
i = Index of worm thread (1 for single, 2 for double, etc.).
L = Lead of worm thread (inches).
M = Revolutions of gear shaft per minute.
M_w = Revolutions of worm shaft per minute.
N = Number of teeth in gear.
P^1 = Circular pitch = Pitch of worm thread (inches).
R = Radius of pitch circle of worm gear (inches).
s = Addendum of tooth (inches).
T = Twisting moment on gear shaft (inch-lbs.).
T_w = Twisting moment on worm shaft (inch-lbs.).
t = Thickness of tooth at pitch line (inches).
W = Load at pitch line (lbs.).

ANALYSIS. The simplest way of analyzing the case of the worm and worm gear is to base it upon an ordinary screw and nut. Take, for example, the lead screw of a common lathe. The carriage carries a nut, through which the lead screw passes. By the rotation of the screw, the carriage, being constrained by the guides to travel lengthwise of the ways, is moved. This motion is, for a single-threaded screw, a distance per revolution equal to the lead of the screw.

Now, suppose that the carriage, instead of sliding along the ways, is compelled to turn about an axis at some point below the ways. Also, suppose the top of the nut to be cut off, and its length made endless by wrapping it around a circle struck from the center about which the carriage rotates. This reduces the nut to a peculiar kind of spur gear, the partial threads of the nut now having the appearance of twisted teeth.

This special form of spur gear, based on the idea of a threaded nut, is known as a **worm gear,** and the screw is termed a **worm.** The teeth are loaded similarly to those of a spur gear, but with the additional feature of a large amount of sliding along the tooth surfaces. This, of course, means considerable friction; and it is in fact possible to utilize the worm and worm gear as an efficient device, only by running the teeth constantly in a bath of oil. Even then the pressures have to be kept well down to insure the required term of life of the tooth surfaces.

It is evident that for one revolution of a single-threaded worm, one tooth of the gear will be passed. The speed ratio between the worm gear and worm shaft will then be equal to the number of teeth in the gear, which is relatively great. Hence the worm and worm gear are principally useful in giving large speed reduction in a small amount of space.

THEORY. The theory of worm-wheel teeth is complicated and obscure. The production of the teeth is simple, a dummy worm with cutting edges, called a "hob," being allowed to carve its way into the worm-gear blank, thus producing the teeth and at the same time driving the worm gear about its axis.

It is clear that if we know the torsional moment on the worm-gear shaft, and the pitch radius of the worm gear, we can find the load on the teeth at the pitch line by dividing the former by the latter. Expressed as an equation:

$$WR = T; \text{or } W = \frac{T}{R}. \qquad (83)$$

How we shall consider this value of W as distributed on the teeth, is a question difficult to answer. The teeth not only are curved to embrace the worm, but are twisted across the face of the gear, so that it would be practically impossible to devise a purely theoretical method of exact calculation. The most reasonable thing to do is to assume the teeth as being equally as strong as spur-gear teeth of the same circular pitch, and to figure them accordingly. It is probably true, however, that the load is carried by more than one tooth, especially in a hobbed wheel; so we shall be safe in assuming that two—and, in case of large wheels, three—teeth divide the load between them. With these considerations borne in mind, the case reduces itself to that of a simple spur-gear tooth calculation, which has already been explained under the heading "Spur Gears."

The worm teeth, or threads, are probably always stronger than the worm-gear teeth; so no calculation for their strength need be made.

The twisting moment on the worm shaft is not determined so directly as in the case of spur gears. The relative number of revolutions of the two shafts depends upon the "lead" of the worm thread and the number of teeth in the gear.

Lead (L) is the distance parallel to the axis of the worm which any point in the thread advances in one revolution of the worm. **Pitch** (P^1) is the distance parallel to the axis of the worm between corresponding points on adjacent threads. The distinction between lead and pitch should be carefully observed, as the two are often confounded, one with the other.

The thread may be single, double, triple, etc., the index of the thread i, being 1, 2, 3, etc., in accordance therewith. The relation between lead and pitch may then be expressed by an equation, thus:

$$L = i\,P^1. \qquad (84)$$

When the index of the thread is changed the speed ratio is changed, the relation being shown by the equation:

$$\frac{M}{M_w} = \frac{i}{N} \qquad (85)$$

If the efficiency were 100 per cent between the two shafts, the twisting moments would be inversely as the ratio of the speeds thus:

$$\frac{T_w}{T} = \frac{M}{M_w} = \frac{i}{N};$$

or,
$$T_w = \frac{Ti}{N}; \qquad (86)$$

but for an efficiency E the equation would be:

$$\frac{T_w}{T} = \frac{i}{EN};$$

or,
$$T_w = \frac{Ti}{EN}. \qquad (87)$$

The diameter of the worm is arbitrary. Change of this diameter has no effect on the speed ratio. It has a slight effect on the efficiency, the smaller worm giving a little higher efficiency. The diameter of the worm runs ordinarily from 3 to 10 times the circular pitch, an average value being $4P^1$ or $5P^1$.

A longitudinal cross-section through the axis of the worm cuts out a rack tooth, and this tooth section is usually made of the standard $14\frac{1}{2}°$ involute form shown in Fig. 46 for a rack.

The end thrust, of a magnitude practically equal to the pressure between the teeth, has to be taken by the hub of the worm against the face of the shaft bearing. A serious loss of efficiency from friction is likely to occur here. This is often reduced, however, by roller or ball bearings. With two worms on the same shaft, each driving into a separate worm gear, it is possible to make one of the worms right-hand thread, and the other left-hand, in which case the thrust is self-contained in the shaft itself, and there is absolutely no end thrust against the face of the bearing. This involves a double outfit throughout, and is not always practicable.

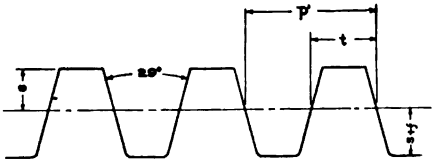


Fig. 46.

There are few mathematical equations necessary for the dimensioning of a worm and worm gear. The formulæ for the tooth parts as given on page 120 apply equally well in this case.

PRACTICAL MODIFICATION. The discussion of the efficiency E of the worm and worm gear is more of a practical than of a theoretical nature. It seems to be true from actual operation, as well as theory, that the steeper the threads the higher the efficiency. In actual practice we seldom have opportunity to change the slope of the thread to get increased efficiency. The slope is usually settled from considerations of speed ratio, or available space, or some other condition. The usual practical problem is to take a given worm and worm gear, and to make out of it as efficient a device as possible. With hobbed gears running in oil baths, and with moderate pressures and speeds, the efficiency will range between 40 per cent and 70 per cent. The latter figure is higher than is usually attained.

To avoid cutting and to secure high efficiency, it seems essential to make the worm and the gear of different materials. The worm-thread surfaces being in contact a greater number of times than the gear teeth, should evidently be of the harder material. Hence we usually find the worm of steel, and the gear of cast iron, brass, or bronze. To save the expense of a large and heavy bronze gear, it is common to make a cast-iron center and bolt a bronze rim to it.

The worm, being the most liable to replacement from wear, it is desirable so to arrange its shaft fastening and general accessibility that it may be readily removed without disturbing the worm gear.

The circular pitch of the gear and the pitch of the worm thread must be the same, and the practical question comes in as to the threads per inch possible to be cut in the lathe in the production of the worm thread. The pitch must satisfy this requirement; hence the pitch will usually be fractional, and the diameter of the worm gear, to give the necessary number of teeth, must be brought to it. While it would perhaps be desirable to keep an even diametral pitch for the worm gear, yet it would be poor design to specify a worm thread which could not be cut in a lathe.

The standard involute of 14½°, and the standard proportions of teeth as given on page 120 are usually used for worm threads. This system requires the gear to have at least 30 teeth, for if fewer teeth are used the thread of the worm will interfere with the flanks of the gear teeth. This is a mathematical relation, and there are methods of preventing it by change of tooth proportions or of angle of worm thread ; but there are few instances in which less than 30 teeth are required, and it is not deemed worth while to go into a lengthy discussion of this point.

The angle of the worm embraced by the worm-gear teeth varies from 60° to 90°, and the general dimensions of rim are made about the same as for spur gears. The arms, or the web, have the same reasons for their size and shape. Probably web gears and cross-shaped arms are more common than oval or elliptical sections.

Worm gears sometimes have cast teeth, but they are for the roughest service only, and give but a point bearing at the middle of the tooth. An accurately hobbed worm gear will give a bearing clear across the face of the tooth, and, if properly set up and cared for, makes a good mechanical device although admittedly of somewhat low efficiency.

Fig. 47 shows a detail drawing of a standard worm and worm gear. It should serve as a suggestion in design, and an illustration of the shop dimensions required for its production.

PROBLEMS ON SPUR, BEVEL, AND WORM GEARS.

1. Calculate proportions of a standard Brown & Sharpe gear tooth of $1\frac{1}{2}$ diametral pitch, making a rough sketch and putting the dimensions on it.

2. Suppose the above tooth to be loaded at the top with 5,000 lbs. If the face be 6 inches, calculate the fiber stress at the pitch line, due to bending.

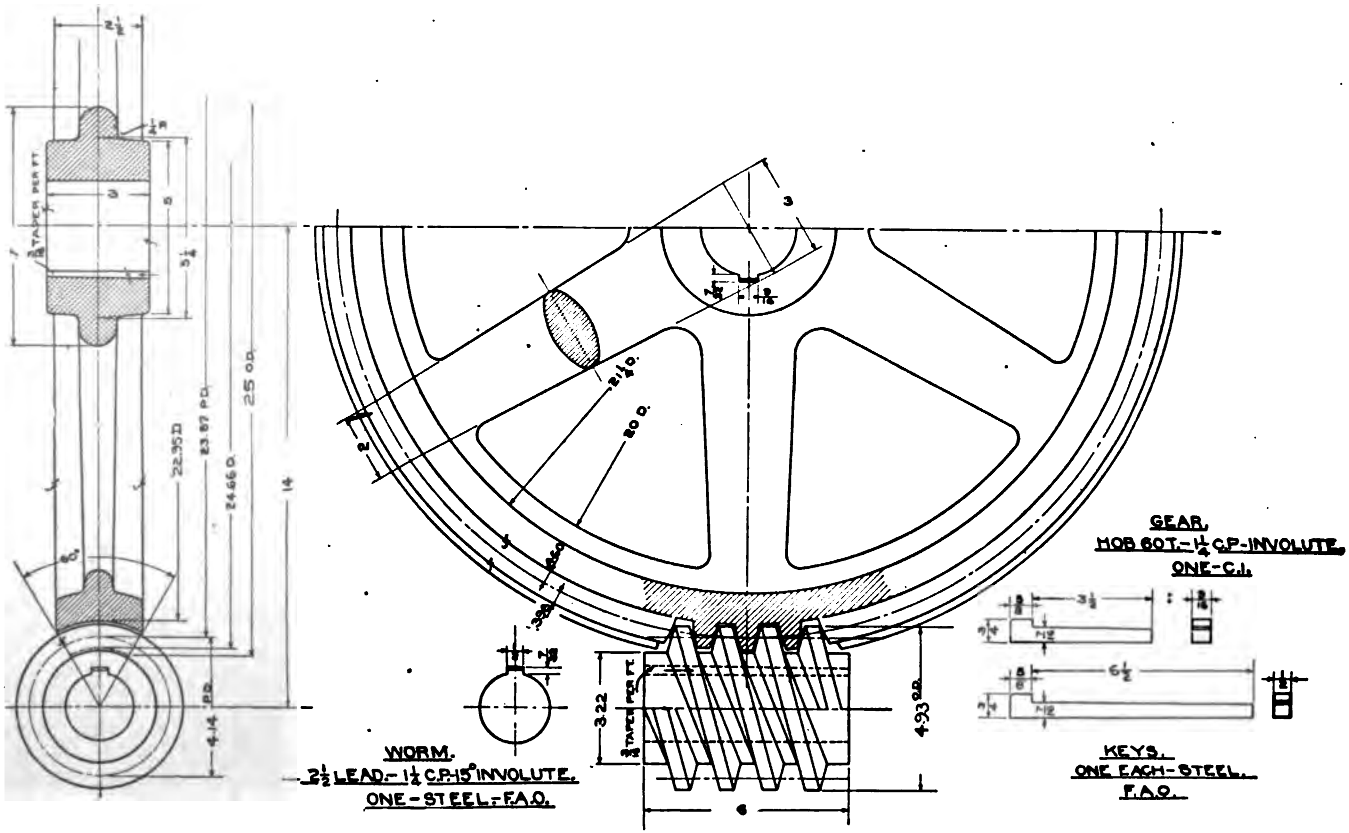
3. A tooth load of 1,200 lbs. is transmitted between two spur gears of 12-inch and 30-inch diameter, the latter gear making 100 revolutions per minute. Calculate a suitable pitch and face of tooth by the "Lewis" formula.

4. Assuming a $\frac{1}{2}$-inch web on the 12-inch gear, calculate the shearing fiber stress at the outside of a hub 4 inches in diameter

5. Design elliptical arms for the 30-inch gear, allowing $S = 2{,}200$.

6. Design cross-shaped arms for 30-inch gear.

7. Calculate the dimensions shown in formulæ 70 to 82 inclusive for a pair of bevel gears of 20 and 60 teeth respectively, 2 diametral pitch, and 4-inch face. (The use of logarithmic tables makes the calculation much easier than with the natural functions.)



SECTION AND ELEVATION OF WORM GEAR.
Fig. 47

8. A worm wheel has 40 teeth, 3 diametral pitch, and double thread. Calculate (*a*) its lead; (*b*) its pitch diameter.

FRICTION CLUTCHES.

NOTATION—The following notation is used throughout the chapter on Friction Clutches:.

a = Angle between clutch face and axis of shaft (degrees)

H = Horse-power (33,000 ft.-lbs. per minute).

μ = Coefficient of friction (per cent).

N = Number of revolutions per minute.

P = Force to hold clutch in gear to produce W (lbs.).

R = Mean radius of friction surface (inches).

T = Twisting moment about shaft axis (inch-lbs.).

V = Force normal to clutch face (lbs.).

W = Load at mean radius of friction surface (lbs.).

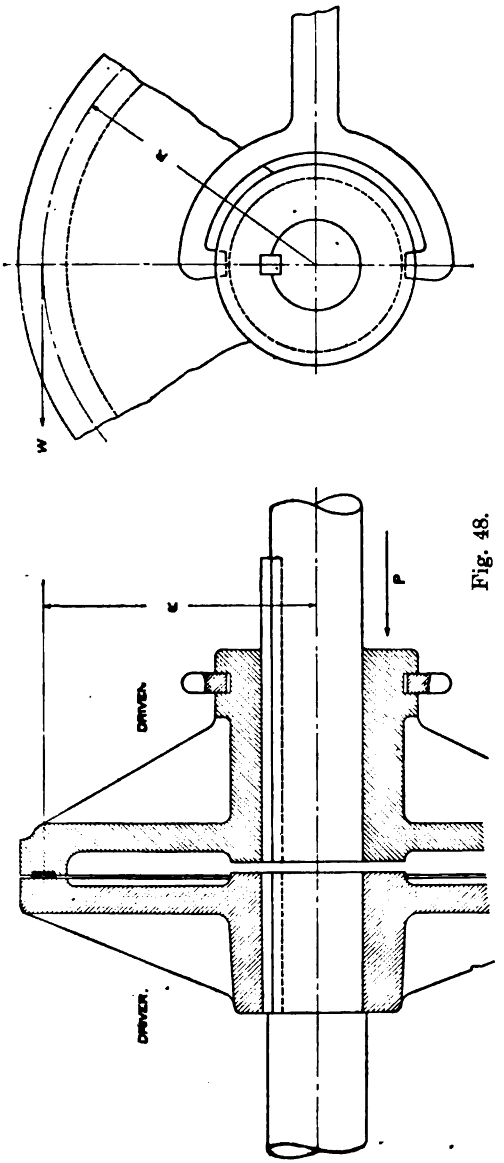


Fig. 48.

ANALYSIS. The friction clutch is a device for connecting at will two separate pieces of shaft, transmitting an amount of power between them to the capacity of the clutch. The connection is usually accomplished while the driving shaft is under full speed, the slipping between the surfaces which occurs during the throwing-in of the clutch, permitting the driven shaft to pick up the speed of the other gradually, without appreciable shock. The disconnection is made in the same manner, the amount of slipping which occurs depending on the suddenness with which the clutch is thrown out.

The force of friction is the sole driving element, hence the problem is to secure as large a force of friction as possible. But friction cannot be secured without a heavy normal pressure between surfaces having a high coefficient of friction between them. The many varieties of friction clutches which are on the market or designed for some special purpose, are all devices for accomplishing one and the same effect, *viz.*, the production of a heavy normal force or pressure between surfaces at such a radius from the driven axis, that the product of the force of friction thereby created and the radius shall equal the desired twisting moment about that axis.

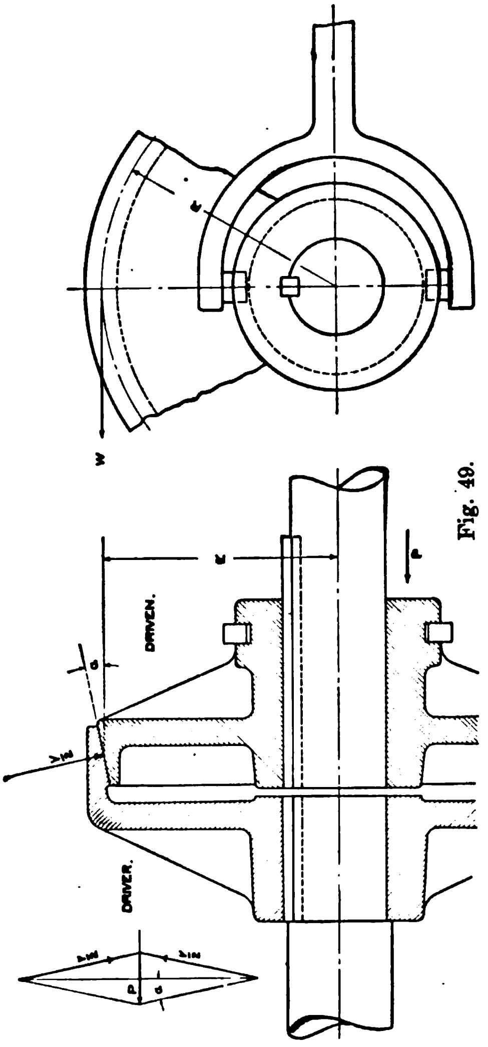


Fig. 49.

Three typical methods of accomplishing this are shown in Figs. 48, 49, and 50. None of these drawings is worked out in operative detail. They are merely illustrations of principle, and are drawn in the simplest form for that purpose.

In Fig. 48 the normal pressure is created in the simplest possible way, an absolutely direct push being exerted between the discs, due to the thrust P of the clutch fork.

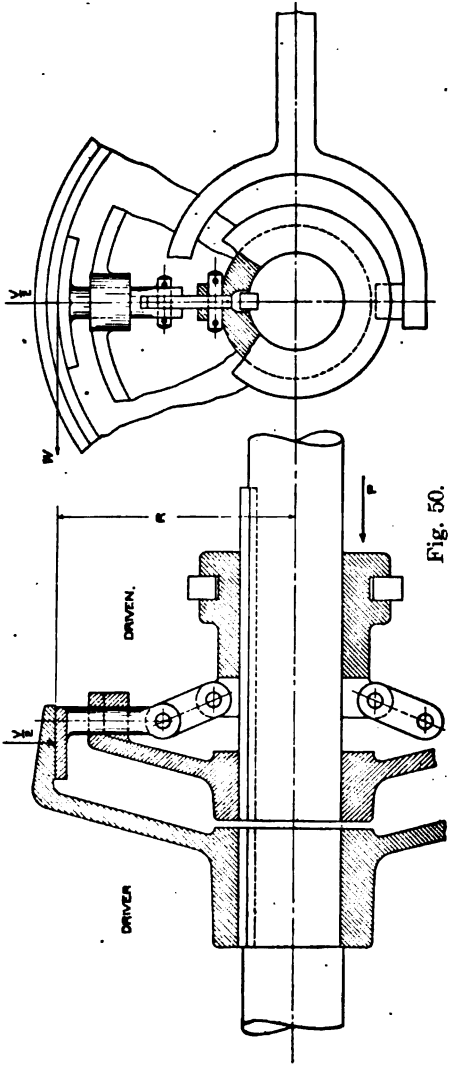


Fig. 50.

In Fig. 49 advantage is taken of the wedge action of the inclined faces, the result being that it takes less thrust P to produce the required normal pressure at the radius R.

In Fig. 50 the inclination of the faces is carried so far that the angle *a* of Fig. 49 has become zero; and by the toggle-joint action of the link pivoted to the clutch collar, the normal force produced may be very great for a slight thrust P. By careful adjustment of the length of the link so that the jaw takes hold of the clutch surface, when the link stands nearly vertical, a very easy operating device is secured, and the thrust P is made a minimum.

THEORY. Referring to Fig. 48 in order to calculate the twisting moment, we must remember that the force of friction between two surfaces is equal to the normal pressure times the coefficient of friction. This, in the form of an equation, using the symbols of the figure, is :

$$W = \mu P. \tag{88}$$

Hence we may consider that we have a force of magnitude μP acting at the mean radius R of the clutch surface. The twisting moment will then be:

$$T = WR = \mu PR. \qquad (89)$$

Referring to equation 54, which gives twisting moment in terms of horse-power, and putting the two expressions equal to each other, we have:

$$T = \frac{63{,}025H}{N} = \mu PR;$$

or,

$$H = \frac{\mu NPR}{63{,}025}. \qquad (90)$$

This expression gives at once the horse-power that the clutch will transmit with a given end thrust P.

In Fig. 49 the equilibrium of the forces is shown in the little sketch at the left of the figure. The clutch faces are supposed to be in gear, and the extra force necessary to slide the two together is not considered, as it is of small importance. The static equations then are:

$$P = 2\frac{V}{2}\sin a;$$

or,

$$V = P \operatorname{cosec} a. \qquad (91)$$

$$W = \mu V = \mu P \operatorname{cosec} a. \qquad (92)$$

$$T = WR = \mu PR \operatorname{cosec} a. \qquad (93)$$

$$T = \frac{63{,}025H}{N} = \mu PR \operatorname{cosec} a;$$

or,

$$H = \frac{\mu NPR \operatorname{cosec} a}{63{,}025}. \qquad (94)$$

In Fig. 50, P would of course be variable, depending on the inclination of the little link. The amount of horse-power which this clutch would transmit would be the same as in the case of the device illustrated in Fig. 49, for an equal normal force V produced.

The further theoretical design of such clutches should be in accordance with the same principles as for arms and webs of pulleys, gears, etc. The length of the hubs must be liberal in order to prevent tipping on the shaft as a result of uneven wear. The end thrust is apt to be considerable; and extra side stiffness must be provided, as well as a rim that will not spring under the radial pressure.

PRACTICAL MODIFICATION. It is desirable to make the most complicated part of a friction clutch the driven part, for then the mechanism requiring the closest attention and adjustment may be brought to and kept at rest when no transmission of power is desired.

Simplicity is an important practical requirement in clutches. The wearing surfaces are subjected to severe usage; and it is essential that they be made not only strong in the first place, but also capable of being readily replaced when worn out, as they are sure to be after some service.

The form of clutch shown in Fig. 50 is the most efficient form of the three shown, although its commercial design is considerably different from that indicated. Usually the jaws grip both sides of the rim, pinching it between them. This relieves the clutch rim of the radial unbalanced thrust.

Adjusting screws must be provided for taking up the wear, and lock nuts for maintaining their position.

Theoretically, the rubbing surfaces should be of those materials whose coefficient of friction is the highest; but the practical question of wear comes in, and hence we usually find both surfaces of metal, cast iron being most common. For metal on metal the coefficient of friction μ cannot be safely assumed at more than 15 per cent, because the surfaces are sure to get oily.

A leather facing on one of the surfaces gives good results as to coefficient of friction, μ having a value, even for oily leather, of 20 per cent. Much slipping, however, is apt to burn the leather; and this is most likely to occur at high speeds.

Wood on cast iron gives a little higher coefficient of friction for an oily surface than metal on metal. Wood blocks can be so set into the face of the jaws as to be readily replaced when worn, and in such case make an excellent facing.

The angle α of a cone friction clutch of the type shown in Fig. 49, may evidently be made so small that the two parts will wedge together tightly with a very slight pressure P; or it may

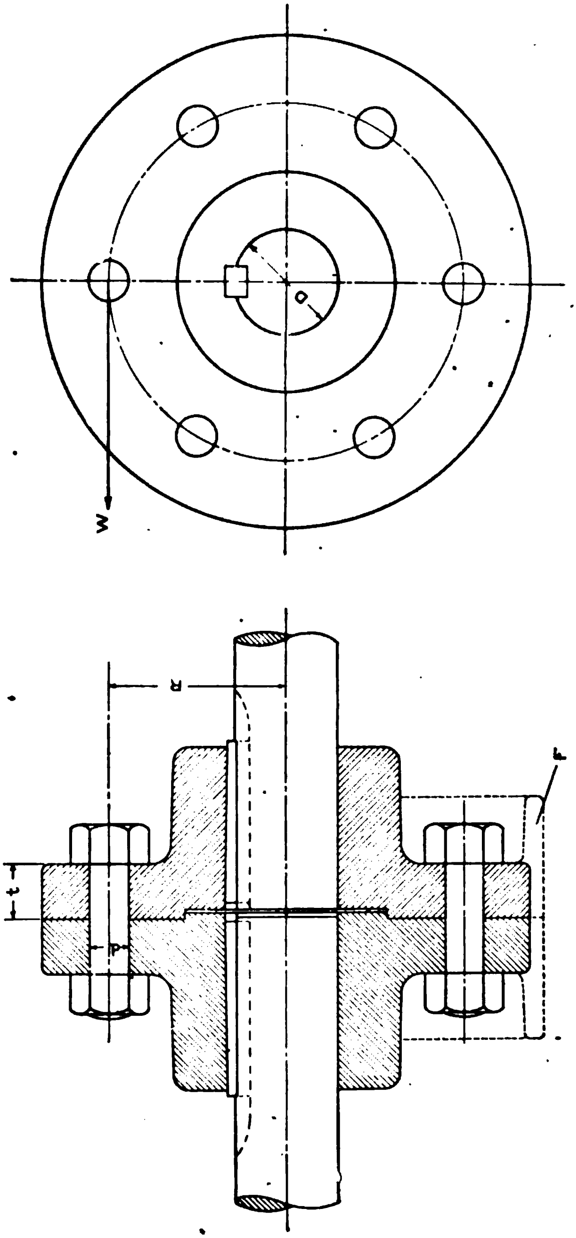


Fig. 51.

be so large as to have little wedging action, and approach the condition illustrated in Fig. 48. Between these limits there is a practical value which neither gives a wedging action so great as to make the surfaces difficult to pull apart, nor, on the other hand, requires an objectionable end thrust along the shaft in order to make the clutch drive properly.

For $a =$ about 15°, the surfaces will free themselves when P is relieved.
" $a =$ " 12°, " " require slight pull to be freed.
" $a =$ " 10°, " " cannot be freed by direct pull of the hand, but require some leverage to produce the necessary force P.

PROBLEMS ON FRICTION CLUTCHES.

1. With what force must we hold a friction clutch in to transmit 30 horse-power at 200 revolutions per minute, assuming working radius of clutch to be 12 inches; coefficient of friction 15 per cent; angle $a = 10^\circ$?

2. How much horse-power could be transmitted, other conditions remaining the same, if the working radius were increased to 18 inches?

3. What force would be necessary in problem 1, if the angle a were 15°, other conditions remaining the same?

COUPLINGS.

NOTATION.—The following notation is used throughout the chapter on Couplings:

D = Diameter of shaft (inches).
d = Diameter of bolt body (inches).
n = Number of bolts.
R = Radius of bolt circle (inches).
S = Safe shearing fiber stress (lbs. per sq. in.).
S_c = Safe crushing fiber stress (lbs. per sq. in.).
T = Twisting moment (inch-lbs.).
t = Thickness of flange (inches).
W = Load on bolts (lbs.).

ANALYSIS. Rigid couplings are intended to make the shafts which they connect act as a solid, continuous shaft. In order that the shaft may be worked up to its full strength capacity, the coupling must be as strong in all respects as the shaft, or, in other words, it must transmit the same torsional moment. In the analysis of the forces which come upon these couplings, it is not considered that they are to take any side load, but that they are to act purely as torsional elements. It is doubtless true that in many cases they do have to provide some side strength and stiffness, but this is not their natural function, nor the one upon which their design is based.

Referring to Fig. 51, which is the type most convenient for analysis, we have an example of the simplest form of **flange coupling.** It consists merely of hubs keyed to the two portions, with flanges driving through shear on a series of bolts arranged concentrically about the shaft. The hubs, keys, and flanges are subject to the same conditions of design as the hubs, keys, and web of a gear or pulley, the key tending to shear and be crushed in the hub and shaft, and the hub tending to be torn or sheared from the flange. The driving bolts, which must be carefully fitted in reamed holes, are subject to a purely shearing stress over their full area at the joint, and at the same time tend to crush the metal in the flange, against which they bear, over their projected area. This latter stress is seldom of importance, the thickness of the flange, for practical reasons, being sufficient to make the crushing stress very low.

THEORY. The theory of hubs, keys, and flanges, being like that already given for pulleys and gears, need not be repeated for couplings. The shearing stress on the bolts is the only new point to be studied.

In Fig. 51, for a twisting moment on the shaft of T, the load at the bolt circle is $W = \frac{T}{R}$. If the number of bolts be n, equating the external force to the internal strength; we have:

$$W = \frac{T}{R} = \frac{S\pi d^2}{4} n. \qquad (95)$$

Although the crushing will seldom be of importance, yet for the sake of completeness its equation is given, thus:

$$W = \frac{T}{R} = S_c d t n. \qquad (96)$$

The internal moment of resistance of the shaft is $\frac{SD^3}{5.1}$; hence the equation representing full equality of strength between the shaft and the coupling, depending upon the shearing strength of the bolts, is:

$$\frac{SD^3}{5.1R} = \frac{S\pi d^2}{4} n. \qquad (97)$$

The theory of the other types of couplings is obscure, except as regards the proportions of the key, which are the same in all cases. The shell of the clamp coupling, Fig. 52, should be thick enough to give equal torsional strength with the shaft; but the exact function which the bolts perform is difficult to determine. In general the bolts clamp the coupling tightly on the shaft and provide rigidity, but the key does the principal amount of the driving. The bolt sizes, in these couplings, are based on judgment and relation to surrounding parts, rather than on theory.

PRACTICAL MODIFICATION. All couplings must be made with care and nicely fitted, for their tendency, otherwise, is to spring the shafts out of line. In the case of the flange coupling, the two halves may be keyed in place on the shafts, the latter then swung on centers in the lathe, and the joint faced off. Thus the joint will be true to the axis of the shaft; and, when it is clamped in position by the bolts, no springing out of line can take place.

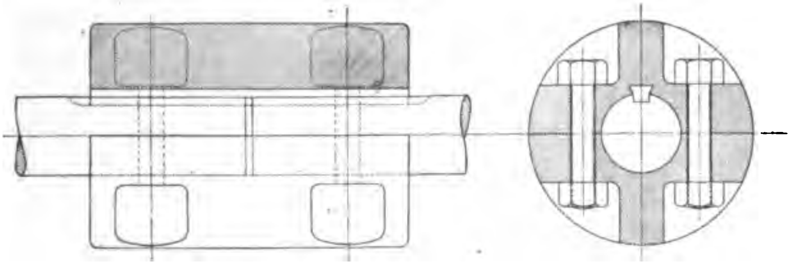


Fig. 52.

A flange F (see Fig. 51) is sometimes made on this form of coupling, in order to guard the bolts. It may be used, also, to take a light belt for driving machinery; but a side load is thereby thrown on the shaft at the joint, which is at the very point where it is desirable to avoid it.

The simplest form of rigid coupling consists of a plain sleeve slipped over from one shaft to the other, when the second is butted up against the first. This is known as a **muff coupling.** When once in place, this is a very excellent coupling, as it is perfectly smooth on the outside, and consists of the fewest possible parts, merely a sleeve and a key. It is, however, expensive to fit, difficult to remove, and requires an extra space of half its length on the shaft over which to be slipped back.

The **clamp coupling** is a good coupling for moderate-sized shafts, where the flange type of Fig. 51 would be unnecessarily expensive. The clamp coupling, Fig. 52, is simply a muff coupling split in halves, and recessed for bolts. It is cheap and is easily applied and removed, even with a crowded shaft. If bored with a piece of paper in the joint, when it is clamped in position it will pinch the shaft tightly and make a rigid connection. It is desirable to have the bolt-heads protected as much as possible, and this may be accomplished by making the outside diameter large enough so that the bolts will not project. Often an additional shell is provided to encase the coupling completely after it is located.

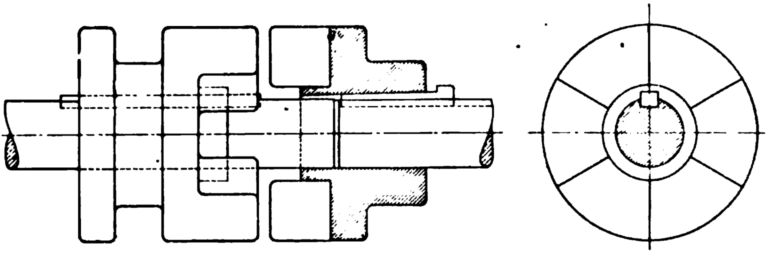


Fig. 53.

There are many other special forms of couplings, some of them adjustable. Most of them depend upon a wedging action exerted by taper cones, screws, or keys. Trade catalogues are to be sought for their description.

The **claw coupling,** Fig. 53, is nothing but a heavy flange coupling with interlocking claws or jaws on the faces of the flanges, to take the place of the driving bolts. This coupling can be thrown in or out as desired, although it usually performs the service of a rigid coupling, as it is not suited to clutching-in during rapid motion, like a friction clutch.

Flexible couplings, which allow slight lack of alignment, are made by introducing between the flanges of a coupling a flexible disc, the one flange being fastened to the inner circle of the disc, the other to the outer circle. This is also accomplished by providing the faces of the flange coupling with pins that drive by pressure together or through leather straps wrapped round the pins. These devices are mostly of a special and often uncertain nature, lacking the positiveness which is one essential feature of a good coupling.

PROBLEMS ON COUPLINGS.

1. A flange coupling of the type of Fig. 51 is used on a shaft 2 inches in diameter. The hub is 3 inches long, and carries a standard key, of proportions indicated below in the table of "Proportions for Gib Keys" (page 166). The bolt circle is 7 inches in diameter, and it is desired to use $\frac{5}{8}$-inch bolts. How many bolts are needed to transmit 60,000 inch-lbs., for a fiber stress in the bolt of 6,000?

2. Using 6 bolts, what diameter of bolt would be required?

3. If four $\frac{3}{4}$-inch bolts were used on a circle of 8 inches diameter, what diameter of shaft would be used in the coupling to give equal strength with the bolts?

BOLTS, STUDS, NUTS, AND SCREWS.

NOTATION—The following notation is used throughout the chapter on Bolts, Studs, Nuts, and Screws:

d = Diameter of bolt (inches).
d_1 = Diameter at root of thread (inches).
H = Height of nut (inches).
I = Initial axial tension (lbs.).
k = Allowable bearing pressure on surface of thread (lbs. per sq. in.).
L = Lead, or distance nut advances along axis in one revolution (inches).
l = Length of wrench handle (inches)
n = Number of threads in nut $= \frac{H}{p}$.
P = Axial load (lbs.).
p = Pitch of thread, or distance between similar points on adjacent threads, measured parallel to axis (inches).
S = Fiber stress (lbs. per sq. in.).
W = Load on bolt (lbs.).

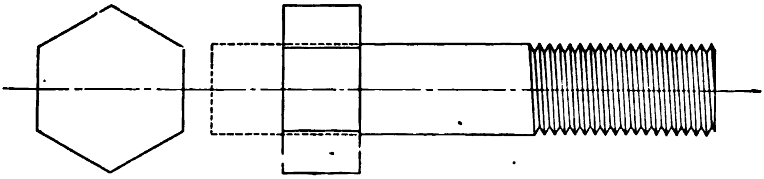


Fig. 54.

ANALYSIS. A **bolt** is simply a cylindrical bar of metal upset at one end to form a head, and having a thread at the other end, Fig. 54. A **stud** is a bolt in which the head is replaced by a thread; or it is a cylindrical bar threaded at both ends, usually having a small plain portion in the middle, Fig. 55. The object of bolts and studs is to clamp machine parts together, and yet permit these same parts to be readily disconnected. The bolt passes through the pieces to be connected, and, when tightened, causes surface compression between the parts, while the reactions on the head and nut produce tension in the bolt. Studs and tap bolts pass through one of the connected parts and are screwed into the other, the stud remaining in position when the parts are disconnected.

As all materials are elastic within certain limits, the action of

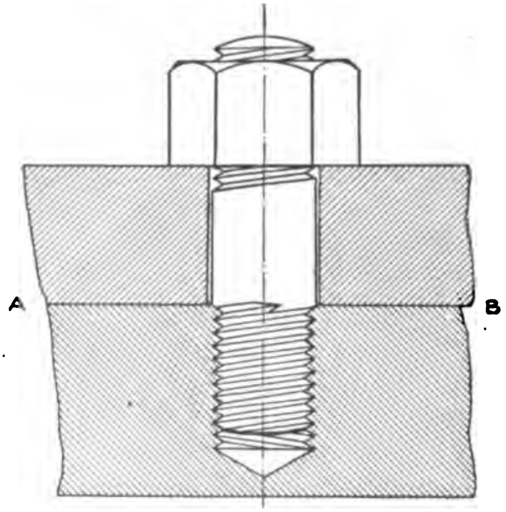


Fig. 55.

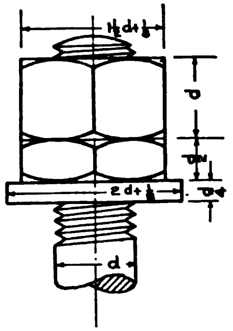


Fig. 55*a*.

a bolt in clamping two machine parts together, more especially if there is an elastic packing between them, may be represented diagrammatically by Fig. 56, in which a spring has been introduced to take the compression due to screwing up the nut. Evidently the tension in the bolt is equal to the force necessary to compress the spring. Now, suppose that two weights, each equal to ½ W, are placed symmetrically on either side of the bolt, then the tension in the bolt will be increased by the added weights if the bolt is perfectly rigid. The bolt, however, stretches; hence some of the compression on the spring is relieved and the total tension in the bolt is less than W + I, by an amount depending on the relative elasticity of the bolt and spring.

Suppose that the stud in Fig. 55 is one of the studs connecting the cover to the cylinder of a steam engine, and that the studs have a small initial tension; then the pressure of the steam loads each stud, and, if the studs stretch enough to relieve the initial pressure between the two surfaces, then their stress is due to the steam pressure only; or, from Fig. 56, when I = W; the initial pressure due to the elasticity of the joint is entirely relieved by the assumed stretch of the studs. Except to prevent leakage, it is seldom necessary to consider the initial tension, for the stretch of the bolt may be counted on to relieve this force, and the working tension on the bolt is simply the load applied.

For shocks or blows, as in the case of the bolts found on the marine type of connecting-rod end, the stretch of the bolts acts like a spring to reduce the resulting tensions. So important is this feature that the body of the bolt is frequently turned down to the diameter of the bottom of the thread, thus uniformly distributing the stretch through the full length of the bolt, instead of localizing it at the threaded parts.

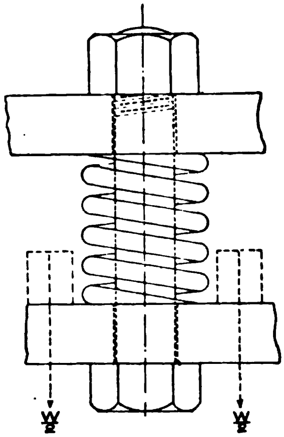


Fig. 56.

In tightening up a bolt, the friction at the surface of the thread produces a twisting moment, which increases the stress in the bolts, just as in the case of shafting under combined tension and torsion; but the increase is small in amount, and may readily be taken care of by permitting low values only for the fiber stress.

In a flange coupling, bolts are acted upon by forces perpendicular to the axis, and hence are under pure shearing stress. If the torque on the shaft becomes too great, failure will occur by the bolts shearing off at the joint of the coupling.

A bolt under tension communicates its load to the nut through the locking of the threads together. If the nut is thin, and the number of threads to take the load few, the threads may break or shear off at the root. With a V thread there is produced a component force, perpendicular to the axis of the bolt, which tends to split the nut.

In screws for continuous transmission of motion and power, the thread may be compared to a rough inclined plane, on which a small block, the nut, is being pushed upward by a force parallel to the base of the plane. The angle at the bottom of the plane is the angle of the helix, or an angle whose tangent is the lead divided by the circumference of the screw. The horizontal force corresponds to the tangential force on the screw. The friction at the surface of the thread produces a twisting moment about the axis of the screw, which, combined with the axial load, subjects the screw to combined tension and torsion. Screws with square threads are generally used for this service, the sides of the thread exerting no bursting pressure on the nut. The proportions of screw thread for transmission of power depend more on the bearing pressure than on strength. If the bearing surface be too small and lubrication poor, the screw will cut and wear rapidly.

THEORY. · A direct tensile stress is induced in a bolt when it carries a load exerted along its axis. This load must be taken by the section of the bolt at the bottom of the thread. If the area at the root of the thread is $\frac{\pi d_1^2}{4}$, and if S is the allowable stress per square inch, then the internal resistance of the bolt is $\frac{S\pi d_1^2}{4}$. Equating the external load to the internal strength we have:

$$W = \frac{S\pi d_1^2}{4}. \qquad (98)$$

For bolts which are used to clamp two machine parts together so that they will not separate under the action of an applied load, the initial tension of the bolt must be at least equal to the applied load. If the applied load is W, then the parts are just about to separate when $I = W$. Therefore the above relation for strength is applicable. As the initial tension to prevent separation should be a little greater than W, a value of S should be chosen so that there will be a margin of safety. For ordinary wrought iron and steel, S may be taken at 6,000 to 8,000.

If, however, the joints must be such that there is no leakage between the surfaces, as in the case of a steam cylinder head, and supposing that elastic packings are placed in the joints, then a much larger margin should be made, for the maximum load which may come on the bolt is I + W, where W is the proportional share of the internal pressure carried by the bolt. In such cases S = 3,000 to 5,000, using the lower value for bolts of less than ¾-inch diameter.

The table given on page 154 will be found very useful in proportioning bolts with U. S. standard thread for any desired fiber stress.

To find the initial tension due to screwing up the nut, we may assume the length of the handle of an ordinary wrench, measured from the center of the bolt, as about 16 times the diameter of the bolt. For one turn of the wrench a force F at the handle would pass over a distance $2\pi l$, and the work done is equal to the product of the force and space, or $F \times 2\pi l$. At the same time the axial load P would be moved a distance p along the axis. Assuming that there is no friction, the equation for the equality of the work at the handle and at the screw is:

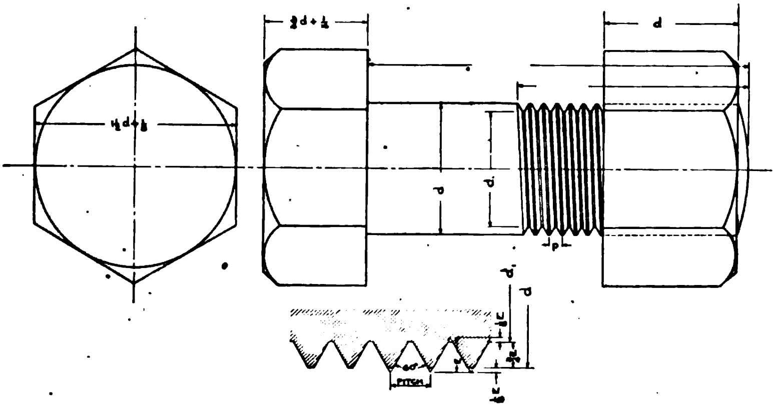


Fig. 56a.

$$F2\pi l = Pp. \qquad (99)$$

Friction, however, is always present; hence the ratio of the useful work (Pp) to the work applied ($F2\pi l$) is not unity as above re-

TABLE FOR STRENGTH OF BOLTS.—U. S. Standard Thread.

BOLT.		DIAMETERS.		AREAS.		APPROXIMATE TENSILE STRENGTH. AT BOTTOM OF THREAD. (IN HUNDREDS OF POUNDS.)					APPROXIMATE SHEARING STRENGTH. FULL BOLT DIAMETER. (IN HUNDREDS OF POUNDS.)			
Diameter. Inches.	Threads per inch.	Bottom of Thread.	Tap Drill.	Bolt Body.	Bottom of Thread.	At 4,000 lbs. per sq. in.	At 5,000 lbs. per sq. in.	At 6,000 lbs. per sq. in.	At 7,000 lbs. per sq. in.	At 10,000 lbs. per. sq. in.	At 4,000 lbs. per sq. in.	At 5,000 lbs. per sq. in.	At 6,000 lbs. per sq. in.	At 7,000 lbs. per sq. in.
¼	20	.18	3/16	.05	.03	1.08	1.35	1.60	1.88	2.69	1.96	2.45	2.95	3.43
5/16	18	.24	¼	.08	.04	1.82	2.27	2.72	3.18	4.54	3.04	3.83	4.60	5.37
⅜	16	.29	5/16	.11	.07	2.71	3.39	4.07	4.75	6.78	4.40	5.52	6.62	7.73
7/16	14	.34	23/64	.15	.09	3.73	4.67	5.60	6.53	9.33	6.00	7.51	9.02	11.00
½	13	.40	13/32	.20	.13	5.00	6.25	7.50	8.75	12.00	7.84	9.81	12.00	14
9/16	12	.45	15/32	.25	.16	6.48	8.10	9.62	11.00	16	9.92	12.00	15	17
⅝	11	.51	17/32	.31	.20	8.04	10.00	12.00	14	20	12.00	15	18	21
¾	10	.62	⅝	.44	.30	12.00	15	18	21	30	18	22	26	31
⅞	9	.73	¾	.60	.42	17	24	25	29	42	24	30	36	42
1	8	.84	27/32	.78	.55	22	27	33	38	55	31	39	47	55
1⅛	7	.94	31/32	.99	.69	28	34	41	48	69	40	50	60	70
1¼	7	1.06	1 3/32	1.23	.89	31	39	47	55	78	49	61	74	86
1⅜	6	1.16	1 3/16	1.48	1.05	42	53	64	72	106	59	74	89	104
1½	6	1.28	1 9/32	1.77	1.29	51	64	77	90	128	71	88	106	124
1⅝	5½	1.39	1 13/32	2.07	1.51	61	78	92	109	153	83	104	124	145
1¾	5	1.49	1½	2.40	1.74	70	88	106	123	176	96	120	144	168
1⅞	5	1.61	1⅝	2.76	2.05	81	101	122	142	203	110	138	166	193
2	4½	1.71	1¾	3.14	2.30	92	115	138	161	230	126	157	186	220
2¼	4½	1.96	1 31/32	3.98	3.02	125	156	187	218	312	159	199	238	278
2½	4	2.17	2 3/16	4.91	3.72	148	185	222	259	370	196	245	294	344
2¾	4	2.42	2 7/16	5.94	4.62	184	230	276	322	460	237	297	356	416
3	3½	2.63	2⅝	7.07	5.43	218	272	326	381	544	283	353	424	495
3¼	3½	2.88	2 29/32	8.29	6.51	264	330	396	462	660	332	415	498	581
3½	3¼	3.10	3 1/32	9.62	7.55	302	376	452	528	754	385	481	577	673
3¾	3	3.32	3 11/32	11.04	8.64	344	430	516	602	860	442	552	663	773
4	3	3.57	3 19/32	12.57	9.99	396	495	594	693	990	503	628	754	880
4¼	2⅞	3.80	3 13/16	14.19	11.33	452	565	678	791	1,130	567	709	851	993
4½	2¾	4.03	4 1/32	15.90	12.74	507	634	760	888	1,268	636	795	951	1,113
4¾	2⅝	4.25	4 9/32	17.72	14.22	567	709	851	1,993	1,420	709	886	1,063	1,240
5	2½	4.48	4½	19.63	15.76	630	788	946	1,103	1,676	785	982	1,178	1,374
5½	2⅜	4.95	4 31/32	23.76	19.27	770	962	1,154	1,347	1,924	950	1,188	1,425	1,663
6	2¼	5.42	5 7/16	28.27	23.09	923	1,159	1,384	1,617	2,307	1,128	1,414	1,696	1,979

lations assume. From numerous experiments on the friction of screws and nuts, it has been found that the efficiency may be as low as 10 per cent. Introducing the efficiency in above equation, it may be written:

$$\frac{Pp}{F2\pi l} = \frac{1}{10}. \qquad (100)$$

Assuming that 50 pounds is exerted by a workman in

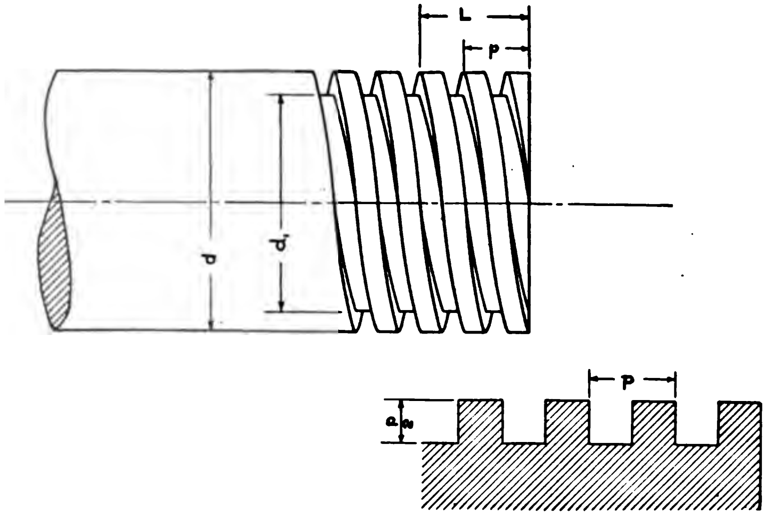


Fig. 58.

tightening up the nut on a 1-inch bolt, the equation above shows that P = 4,021 pounds; or the initial tension is somewhat less than the tabular safe load shown for a 1-inch bolt, with S assumed at 10,000 pounds per sq. inch.

For shearing stresses the bolt should be fitted so that the body of the bolt, not the threads, resists the force tending to shear off the bolt perpendicular to its axis. The internal strength of the bolt to resist shear is the allowable stress S times the area of the bolt in shear, or $\frac{S\pi d^2}{4}$. If W represents the external force tending to shear the bolt the equality of the external force to the internal strength is :

$$W = \frac{S\pi d^2}{4}. \qquad (101)$$

Reference to the table on page 154 for the shearing strength of bolts, may be made to save the labor of calculations.

Let Fig. 58 represent a square thread screw for the transmission of motion. The surface on which the axial pressure bears, if n is the number of threads in a nut, is $\frac{\pi}{4}(d^2 - d_1^2)\,n$. Suppose that a pressure of k pounds per square inch is allowed on the surface of the thread. Then the greatest permissible axial load P must not exceed the allowable pressure; or, equating,

$$P = k\,\frac{\pi}{4}(d^2 - d_1^2)\,n \qquad (102)$$

The value of k varies with the service required. If the motion be slow and the lubrication very good, k may be as high as 900. For rapid motion and doubtful lubrication, k may not be over 200. Between these extremes the designer must use his judgment, remembering that the higher the speed the lower is the allowable bearing pressure.

PRACTICAL MODIFICATION. It will be noted in the formulæ for bolt strengths that different values for S are assumed. This is necessary on account of the uncertain initial stresses which are produced is setting up the nuts. For cases of mere fastening, the safe tension is high, as just before the joint opens the tension is about equal to the load and yet the fastening is secure. On the other hand, bolts or studs fastening joints subjected to internal fluid pressure must be stressed initially to a greater amount than the working pressure which is to come on the bolt. As this initial stress is a matter of judgment on the part of the workman, the designer, in order to be on the safe side, should specify not less than $\frac{5}{8}$-inch or $\frac{3}{4}$-inch bolts for ordinary work, so that the bolts may not be broken off by a careless workman accidentally putting a greater force than necessary on the wrench handle. In making a steam-tight joint, the spacing of the bolts will generally determine their number; hence we often find an excess of bolt strength in joints of this character.

Through bolts are preferred to studs, and studs to tap bolts or cap screws. If possible, the design should be such that through bolts may be used. They are cheapest, are always in standard stock, and well resist rough usage in connecting and disconnecting. The threads in cast iron are weak and have a tendency to crumble; and if a through bolt cannot be used in such a case, a stud, which can be placed in position once for all, should be employed—not a tap bolt, which injures the thread in the casting every time it is removed.

The plain portion of a stud should be screwed up tight against the shoulder, and the tapped hole should be deep enough to prevent bottoming. To avoid breaking off the stud at the shoulder, a neck, or groove, may be made at the lower end of the thread entering the nut.

To withstand shearing forces the bolts must be fitted so that no lost motion may occur, otherwise pure shearing will not be secured.

Nuts are generally made hexagonal, but for rough work are often made square. The hexagonal nut allows the wrench to turn through a smaller angle in tightening up, and is preferred to the square nut. Experiments and calculations show that the height of the nut with standard threads may be about $\frac{1}{2}$ the diameter of the bolt and still have the shearing strength of the thread equal to the tensile strength of the bolt at the root of the thread. Practically, however, it is difficult to apply such a thin wrench as this proportion would call for on ordinary bolts. More commonly the height of the nut is made equal to the diameter of the bolt so that the length of thread will guide the nut on the bolt, give a low bearing pressure on the threads, and enable a suitable wrench to be easily applied. The standard proportions for bolts and nuts may be found in any handbook. Not all manufacturers conform to the United States standard; nor do manufacturers in all cases conform to one another in practice.

If the bolt is subject to vibration, the nuts have a tendency to loosen. A common method of preventing this is to use double nuts, or **lock nuts**, as they are called (see Fig. 55 A). The under nut is screwed tightly against the surface, and held by a wrench while the second nut is screwed down tightly against the first. The effect is to cause the threads of the upper nut to bear against the under sides of the threads of the bolt. The load on the bolt is sustained therefore by the upper nut, which should be the thicker of the two; but for convenience in applying wrenches the position of the nuts is often reversed.

The form of thread adapted to transmitting power is the square thread, which, although giving less bursting pressure on the nut, is not as strong as the V thread for a given length, since the total section of thread at the bottom is only ½ as great. If the pressure is to be transmitted in but one direction, the two types may be combined advantageously to form the buttress thread of the proportions shown in Fig. 59. Often, as in the carriage of a lathe, to allow the split nut to be opened and closed over the lead screw, the sides of the thread are placed at a small angle, say 15°, to each other, as illustrated in Fig. 60.

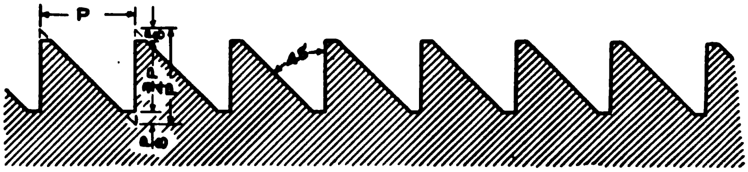


Fig. 59.

The practical commercial forms in which we find screwed fastenings are included in five classes, as follows:

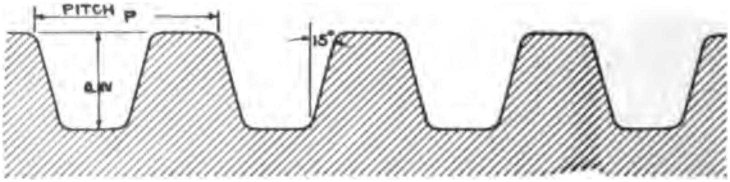


Fig. 60.

1. **Through bolts** (Fig. 61), usually rough stock, with square upset heads, and square or hexagonal nuts.

2. **Tap bolts** (Fig. 62), also called **cap screws.** These usually have hexagonal heads, and are found both in the rough form, and finished from the rolled hexagonal bar in the screw machine.

3. **Studs** (Fig. 63), rough or finished stock, threaded in the screw machine.

4. **Set screws** (Fig. 64), usually with square heads, and case-hardened points. Many varieties of set screws are made, the

principal distinguishing feature of each being in the shape of the point. Thus, in addition to the plain beveled point, we find the "cupped," rounded, conical, and "teat" points.

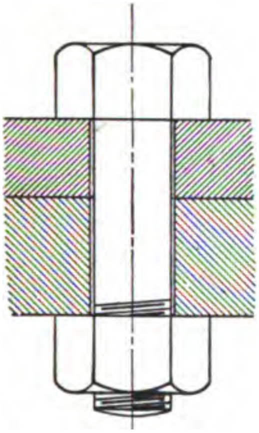


Fig. 61.

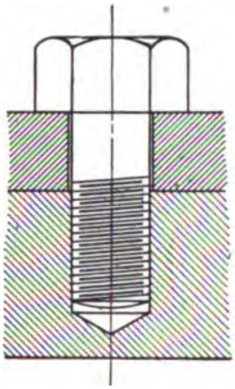


Fig. 62.

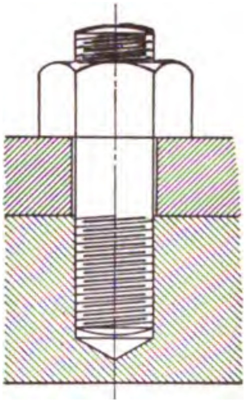


Fig. 63.

5. **Machine screws** (Fig. 64*a*), usually round, "button," or countersunk head. Common proportions are indicated relative to diameter of body of screw.

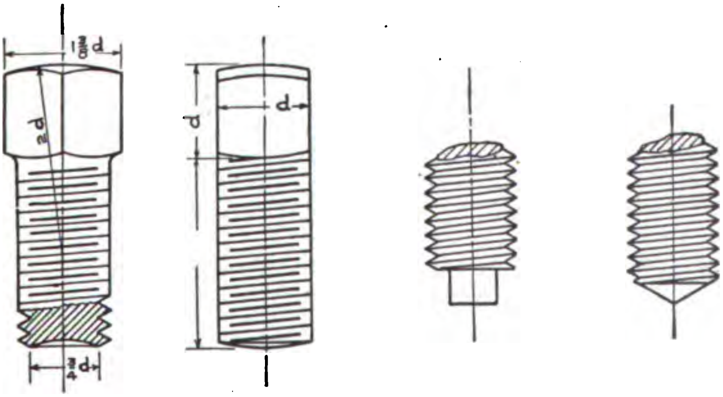


Fig. 64.

PROBLEMS ON BOLTS, STUDS, NUTS, AND SCREWS.

1. Calculate the diameter of a bolt to sustain a load of 5,000 lbs.

2. The shearing force to be resisted by each of the bolts of a flange coupling is 1,200 lbs. What commercial size of bolt is required?

3. With a wrench 16 times the diameter of the bolt, and an efficiency of 10 per cent, what axial load can a man exert on a standard $\frac{3}{4}$-inch bolt, if he pulls 40 lbs. at the end of the wrench handle?

4. A single, square-threaded screw of diameter 2 inches, lead $\frac{1}{4}$ inch, depth of thread $\frac{1}{8}$ inch, length of nut 3 inches, is to be allowed a bearing pressure of 300 lbs. per square inch. What axial load can be carried?

5. Calculate the shearing stress at the root of the thread in problem 4.

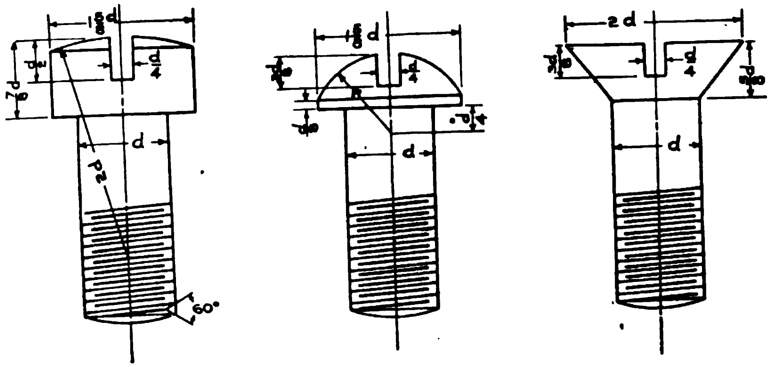


Fig. 64*a*.

KEYS, PINS, AND COTTERS.

NOTATION—The following notation is used throughout the chapter on Keys, Pins, and Cotters:

D = Average diameter of rod (inches).
D_1 = Outside diameter of socket (inches).
d = Diameter of shaft (inches).
L = Length of key (inches).
P = Driving force (lbs.).
P_1 = Axial load on rod (lbs.).
R = Radius at which P acts (inches).
S_c = Safe crushing fiber stress (lbs. per sq. in.).
S_s = Safe shearing fiber stress (lbs. per sq. in.).
S_t = Safe tensile fiber stress (lbs. per sq. in.).
T = Thickness of key (inches).
W = Width of key (inches).
w = Average width of cotter (inches).
w_1 = End of slot to end of rod (inches).
w_2 = End of slot to end of socket (inches).

KEYS AND PINS.

ANALYSIS. Keys and pins are used to prevent relative rotary motion between machine parts intended to act together as one piece. If we drill completely through a hub and across the shaft, and insert a tightly fitted pin, any rotary motion of the one will be transmitted to the other, provided the pin does not fail by shearing off at the joint between the shaft and the hub. The shearing area is the sum of the cross-sections of the pin at the joint.

We may drill a hole in the joint, the axis of the hole being parallel to the axis of the shaft, and drive in a pin, in which case we introduce a shearing area as before, but the area is now equal to the diameter of the pin multiplied by its length, and the pin is stressed sidewise, instead of across. It is evident in the sidewise case that we may increase the shearing area to anything we please, without changing the diameter of the pin, merely by increasing the length of the pin.

As there are some manufacturing reasons why a round pin placed lengthwise in the joint is not always applicable, we may make the pin a rectangular one, in which case it is called a **key.**

When pins are driven across the shaft as in the first instance, they are usually made taper. This is because it is easier to ream a taper hole to size than a straight hole, and a taper pin will drive more easily than a straight pin, it not being necessary to match the hole in hub and shaft so exactly in order that the pin may enter. The taper pin will draw the holes into line as it is driven, and can be backed out readily in removal.

Keys of the rectangular form are either straight or tapered, but for different reasons from those just stated for pins. Straight keys have working bearing only at the sides, driving purely by shear, crushing being exerted by the side of the key in both shaft and hub, over the area against the key. The key itself does not prevent end motion along the shaft; and if end motion is not desired, auxiliary means of some sort must be resorted to, as, for example, set screws through the hub jamming hard against the top of the key.

If end motion along the shaft is desired, the key is called a **spline,** and, while not jammed against the shaft, is yet prevented from changing its relation to the hub by some means such as illustrated in Fig. 65.

Taper keys not only drive through sidewise shearing strength, but prevent endwise motion by the wedging action exerted between the shaft and hub. These keys drive more like a strut from corner to corner; but this action is incidental rather than intentional, and the proportions of a taper key should be such that it will give its full resisting area in shearing and crushing, the same as a straight key.

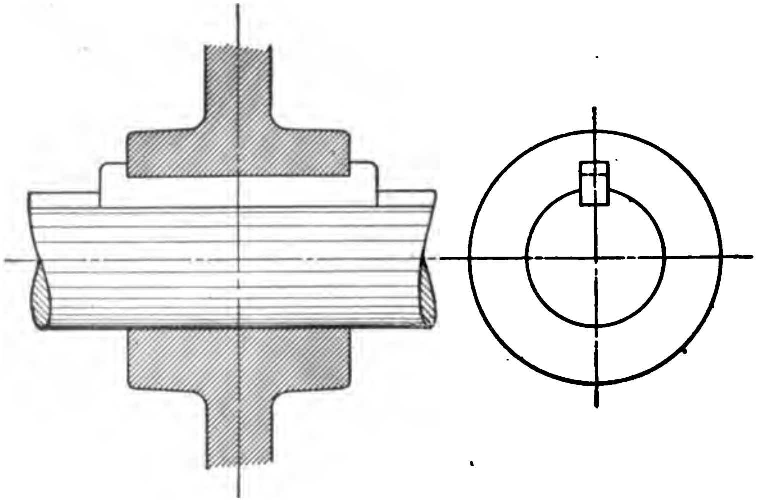


Fig. 65.

THEORY. Suppose that the pin illustrated in Fig. 66 passes through hub and shaft, and the driving force P acts at the radius R; then the force which is exerted at the surface of the shaft to shear off the pin at the points A and B is $\frac{2\ PR}{d}$. If D_1 is the average diameter of the pin, its shearing strength is $\frac{2\pi D_1^2 S_s}{4}$. Equating the external force to the internal strength, we have:

$$\frac{2PR}{d} = \frac{2\pi D_1^2\ S_s}{4};$$

or,

$$D_1 = \sqrt{\frac{4PR}{\pi d S_s}}. \qquad (103)$$

In Fig. 67 a rectangular key is sunk half way in hub and shaft according to usual practice. Here the force at the surface of the shaft, calculated the same as before, not only tends to shear off the key along the line AB, but tends to crush both the portion in the shaft and in the hub. The shearing strength along the line AB is LWS_s. Equating external force to internal strength, we have:

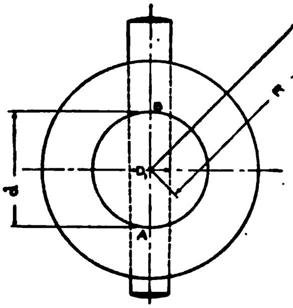


Fig. 66.

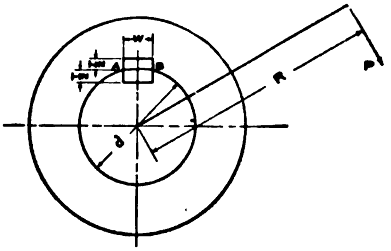


Fig. 67.

$$\frac{2PR}{d} = LWS_s ;$$

or,
$$W = \frac{2PR}{dLS_s}. \qquad (104)$$

The crushing strength is, of course, that due to the weaker metal, whether in shaft or hub. Let S_c be this least safe crushing fiber stress. The crushing strength then is $\frac{LT}{2} S_c$, and, equating external force to internal strength, we have:

$$\frac{2PR}{d} = \frac{LT}{2} S_c ;$$

or,
$$T = \frac{4PR}{dLS_c}. \qquad (105)$$

The proportions of the key must be such that the equations as above, both for shearing and for crushing, shall be satisfied.

PRACTICAL MODIFICATION. Pins across the shaft can be used to drive light work only, for the shearing area cannot be very large. A large pin cuts away too much area of the shaft, decreasing the latter's strength, Pins are useful in preventing end motion, but in this case are expected to take no shear, and may be of small diameter. The common split pin is especially adapted to this service, and is a standard commercial article.

Taper pins are usually listed according to the Morse standard taper, proportions of which may be found in any handbook. It is desirable to use *standard* taper pins in machine construction, as the reamers are a commercial article of accepted value, and readily obtainable in the machine-tool market.

With properly fitted keys, the shearing strength is usually the controlling element. For shafts of ordinary size, the standard proportions as given in tables like that below are safe enough without calculation, up to the limit of torsional strength of the shaft. For special cases of short hubs or heavy loads, a calculation is needed to check the size, and perhaps modify it.

Splines, also known as "feather keys," require thickness greater than regular keys, on account of the sliding at the sides. A table suggesting proportions for splines is given on page 166.

Though the spline may be either in the shaft or hub, it is the more usual thing to find the spline dovetailed (Fig. 67*a*), "gibbed," or otherwise fastened in the hub; and a long spline way made in the shaft, in which it slides.

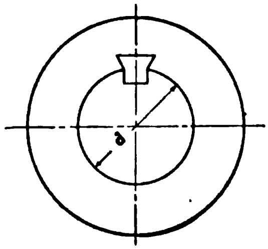


Fig. 67*a*.

The straight key, accurately fitted, is the most desirable fastening device for accurate machines, such as machine tools, on account of the fact that there is absolutely no radial force exerted to throw the parts out of true. It, however, requires a tight fit of hub to shaft, as the key cannot be relied upon to take up any looseness.

The taper key (Fig. 68), by its wedging action, will take up some looseness, but in so doing throws the parts out slightly. Or, even if the bored fit be good, if the taper key be not driven home with care, it will spring the hub, and make the parts run untrue. The great advantage, however, that the taper key has of holding the hub from endwise motion, renders it a very useful and practical article. It is usually provided with a head, or "gib," which permits a draw hook to be used to wedge between the face of the hub and the key to facilitate starting the key from its seat.

Two keys at 90° from each other may be used in cases where one key will not suffice. The fine workmanship involved in spacing these keys so that they will drive equally makes this plan inadvisable except in case of positive and unavoidable necessity.

The "Woodruff" key (Fig. 69) is a useful patented article for certain locations. This key is a half-disc, sunk in the shaft and the hub is slipped over it. A simple rotary cutter is dropped into the shaft to produce the key seat; and on account of the depth in the shaft, the tendency to rock sidewise is eliminated, and the drive is purely by shear.

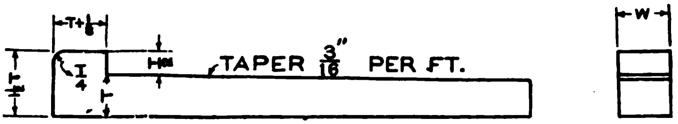


Fig. 68.

Keys may be milled out of solid stock, or drop-forged to within a small fraction of finished size. The drop-forged key is an excellent modern production and requires but a minimum amount of fitting. Any key, no matter how produced, requires some hand fitting and draw filing to bring it properly to its seat and give it full bearing.

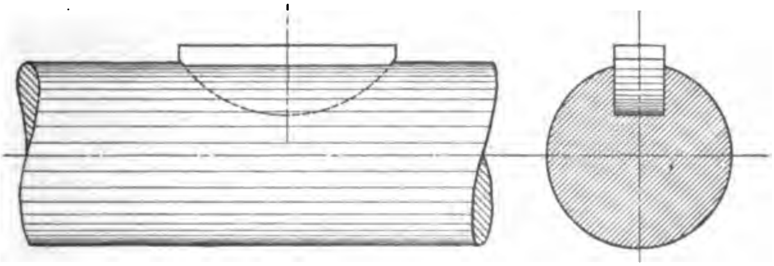


Fig. 69.

It is good mechanical policy to avoid keyed fastenings whenever possible. This does not mean that keys may never be used, but that a key is not an ideal way to produce an absolutely positive drive, partly because it is an expensive device, and partly because the tendency of any key is to work itself loose, even if carefully fitted.

The following tables are suggested as a guide to proportions of gib keys and feather keys, and will be found useful in the absence of any manufacturer's standard list:

Fig. 70. **PROPORTIONS FOR GIB KEYS.**

Diameter of shaft (d), inches.	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{5}{8}$	2	$2\frac{1}{2}$	$3\frac{1}{4}$	4	5	$6\frac{1}{2}$
Width (W), inches.	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{5}{8}$
Thickness (T), inches.	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{17}{32}$	$\frac{21}{32}$	$\frac{13}{16}$	1	$1\frac{1}{4}$

Fig. 71. **PROPORTIONS FOR FEATHER KEYS.**

Diameter of Shaft (d), inches,	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$
Width (W), inches.	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$
Thickness (T), inches.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$

COTTERS.

ANALYSIS. Cotters are used to fasten hubs to rods rather than shafts, the distinction between a rod and a shaft being that a rod takes its load in the direction of its length, and does not drive by rotation. A cotter, therefore, is nothing but a cross-pin of modified form, to take shearing and crushing stress in the direction of the axis of the rod, instead of perpendicular to it.

Referring to Fig. 72, one will see that the cotter is made long and thin—long, in order to get sufficient shearing area to resist shearing along lines A and B; thin, in order to cut as little cross-sectional area out of the body of the shaft as possible. The cotter itself tends to shear along the lines A and B, and crush along the surfaces K, G, and J. The socket tends to crush along the surfaces K and G. The rod end D tends to be sheared out along the lines C H and Q E, and also to be crushed along the surface J. The socket tends to be sheared along the lines V U and X Y.

The cotter is made taper on one side, thus enabling it to draw up the flange of the rod tightly against the head of the socket. This taper must not be great enough to permit easy "backing out" and loosening of the cotter under load or vibration in the rod. In responsible situations this cannot be safely guarded against except through some auxiliary locking device, such as lock nuts on the end of the cotter (Fig. 73).

THEORY. Referring to Fig. 72, assume an axial load of P_1, as shown. The successive equations of external force to internal strength are enumerated below, for the different actions that take place:

For shearing along lines A and B, w being the average width of cotter, and S_s safe shearing stress of cotter,

$$P_1 = 2TwS_s. \qquad \textbf{(106)}$$

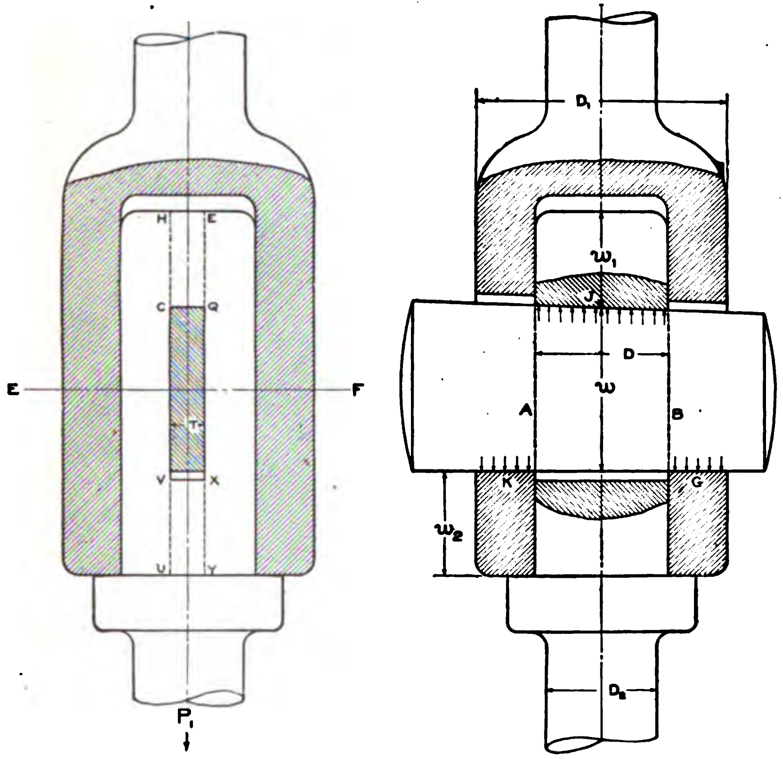


Fig. 72.

For crushing along surfaces K and G, S_c being least safe crushing stress, whether of cotter or socket,

$$P_1 = T(D_1 - D)S_c. \qquad \textbf{(107)}$$

For crushing along surface J, S_c being least safe crushing stress, whether of cotter or socket,

$$P_1 = DTS_c. \qquad \textbf{(108)}$$

For shearing along surfaces CH and QE, S_s being safe shearing stress of rod end, and w_1 end of slot to end of rod,

$$P_1 = 2w_1DS_s. \qquad \textbf{(109)}$$

For tension in rod end at section across slot, S_t being safe tensile stress in rod end,

$$P_1 = (\frac{\pi D^2}{4} - TD)S_t. \qquad \textbf{(110)}$$

For tension in socket at section across slot, S_t being safe tensile stress in socket,

$$P_1 = [\frac{\pi D_1^2}{4} - \frac{\pi D^2}{4} - T(D_1 - D)]S_t. \qquad \textbf{(111)}$$

For shearing in socket along the lines VU and XY, S_s being safe shearing stress in the socket, and w_2 end of slot to end of socket,

$$P_1 = 2w_2(D_1 - D)S_s. \qquad \textbf{(112)}$$

The proportions of cotter and socket may be fixed to some extent by practical or assumed conditions. The dimensions may then be tested by the above equations, that the safe working stresses may not be exceeded, the dimensions being then modified accordingly.

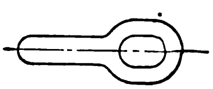
The steel of which both cotter and rod would ordinarily be made has range of working fiber stress as follows :

Tension, 8,000 to 12,000 (lbs. per sq. in.)

Compression, 10,000 to 16,000 (lbs. per sq. in.)

Shear, 6,000 to 10,000 (lbs. per sq. in.)

The socket, if made of cast iron, will be weak as regards tension, tendency to shear out at



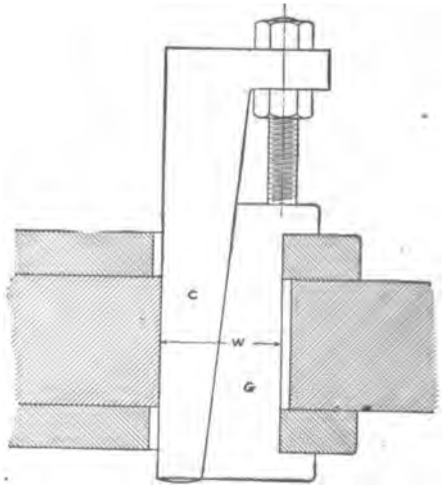


Fig. 73.

the end, and tendency to split. The uncertainty of cast iron to resist these is so great that the hub or socket must be very clumsy in order to have enough surplus strength. This is always a noticeable feature of the cotter type of fastening, and cannot well be avoided.

PRACTICAL MODIFICATION. The driving faces of the cotter are often made semicircular. This not only gives more shearing area at the sides of the slots, but makes the production of the slots easier in the shop. It also avoids the general objection to sharp corners—namely, a tendency to start cracks.

A practicable taper for cotters is $\frac{1}{2}$ inch per foot. This will under ordinary circumstances prevent the cotter from backing out under the action of the load. When set screws against the side of the cotter, or lock nuts are used, as in Fig. 73, the taper may be greater than this, perhaps as much as $1\frac{1}{2}$ inches per foot.

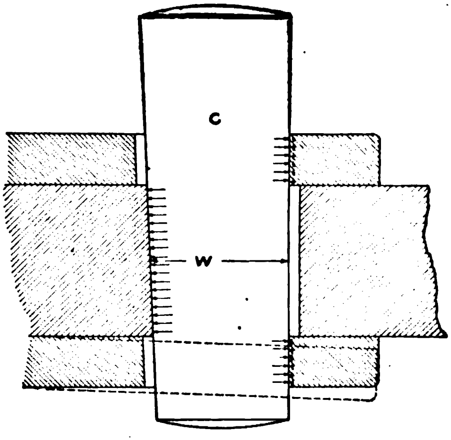


Fig. 74.

In the common use of the cotter for holding the strap at the ends of connecting rods, the strap acts like a modified form of socket. This is shown in Figs. 73 and 74. Here, in addition to holding the strap and rod together lengthwise, it may be necessary to prevent their spreading, and for this purpose an auxiliary piece G with gib ends is used. The tendency without this extra piece is shown by the dotted lines in Fig. 74.

The general mechanical fault with cottered joints is that the action of the load, especially when it constantly reverses, as in pump piston rods, always tends to work the cotter loose. Vibration also tends to produce the same effect. Once this looseness is started in the joint, the cotter loses its pure crushing and shearing action, and begins to partake of the nature of a hammer, and pounds itself and its bearing surfaces out of their true shape. Instead of a collar on the rod, we often find a taper fit of the rod in the socket; and any looseness in this case is still worse, for the rod then has end play in the socket, and by its "shucking" back and forth tends to split open the socket.

The only answer to these objections is to provide a positive locking device, and take up any looseness the instant it appears.

PROBLEMS ON KEYS, PINS, AND COTTERS.

1. Calculate the safe load in shear which can be carried on a key $\frac{1}{2}$ inch wide, $\frac{3}{8}$ inch thick, and 5 inches long. Assume $S_s = 6{,}000$.

2. Assuming the above key to be $\frac{3}{16}$ inch in hub and $\frac{3}{16}$ inch in shaft, test its proportions for crushing, at $S_c = 16{,}000$.

3. A gear 60 inches in diameter has a load of 3,000 lbs. at the pitch line. The shaft is 4 inches in diameter, in a hub, 5 inches long; and the key is a standard gib key as given in the table. Test its proportions for shearing.

4. A piston rod 2 inches in diameter carries a cotter $\frac{3}{8}$ inch thick, and has an axial load of 20,000 lbs. Calculate the average width of the cotter. $S_s = 9{,}000$.

5. Calculate fiber stress in rod in preceding problem at section through slot.

6. How far from the end of rod must the end of slot be?

7. Calculate the crushing fiber stresses on cotter, rod, and socket.

8. How far from the end of socket must the end of slot be, assuming the socket to be of steel?

BEARINGS, BRACKETS, AND STANDS.

NOTATION—The following notation is used throughout the chapters on Bearings Brackets, and Stands.

A = Area (square inches).
a = Distance between bolt centers (inches).
b = Width of bracket base (inches).
c = Distance of neutral axis from outer fiber (inches).
D = Diameter of shaft (inches).
d = Diameter of bolt body (inches).
d_1 = Diameter at root of thread (inches).
H = Horse-power.
h = Thickness of cap at center (inches).
I = Moment of inertia.
L = Length of bearing (inches).
μ = Coefficient of friction (per cent).

N = Number of revolutions per minute.
n = Number of bolts in cap.
n_1 = Number of bolts in bracket base.
P = Total pressure on bearing (lbs.).
p = Pressure per square inch of projected area (lbs).
S = Safe tensile fiber stress (lbs.).
S_s = " shearing " (lbs.).
T = Total load on bolts at top of bracket (lbs.).
t = Thickness of bracket base (inches).
x = Distance from line of action of load to any section of bracket (inches).

ANALYSIS. Machine surfaces taking weight and pressure of other parts in motion upon them are, in general, known as **bearings.** If the motion is rectilinear, the bearing is termed a **slide, guide,** or **way,** such as the cross slide of a lathe, the crosshead guide of a steam engine, or the ways of a lathe bed.

If the motion is a rotary one, like that of the spindle of a lathe, the simple word "bearing" is generally used.

In any bearing, sliding or rotary, there must be strength to carry the load, stiffness to distribute the pressure evenly over the full bearing surface, low intensity of such pressure to prevent the lubricant from being squeezed out and to minimize the wear, and sufficient radiating surface to carry away the heat generated by friction of the surfaces as fast as it is generated. Sliding bearings are of such varied nature, and exist under conditions so peculiar to each case, that a general analysis is practically impossible beyond that given in the sentence above.

Rotary bearings can be more definitely studied, as there are but two variable dimensions, diameter and length, and it is the proper relation between these two that determines a good bearing. The size of the shaft, as noted under "Shafts," is calculated by taking the bending moment at the center of the bearing, combining it with the twisting moment, and solving for the diameter consistent with the assumed fiber stress. But this size must then be tried for deflection due to the bending load, in order that the requirement for stiffness may be fulfilled. When this is accomplished, the friction at the bearing surface may still generate so much heat that the exposed surface of the bearing will not radiate it as fast as generated, in which case the bearing gets hotter and hotter, until it finally burns out the lubricant and melts the lining of the bearing, and ruin results.

The heat condition is usually the critical one, as it is very easy to make a short bearing which is strong enough and amply stiff for the load it carries, but which nevertheless is a failure as a bearing, because it has so small a radiating surface that it cannot run cool.

The side load which causes the friction and the consequent development of heat, is due to the pull of the belt in the case of pulleys, the load on the teeth of gears, the pull on cranks and levers, the weight of parts, etc. If we could exert pure torsion on shafts without any side pressure, and counteract all the weight that comes on the shaft, we should not have any trouble with the development of heat in bearings; in fact, there would theoretically be no need of bearings, as the shafts would naturally spin about their axes, and would not need support.

It can be shown, theoretically, that the radiating surface of a bearing increases relatively to the heat generated by a given side load, *only when the length of the bearing is increased.* In other words, increasing the diameter and not the length, theoretically increases the heat generated per unit of time just as much as it increases the radiating surface; hence nothing is gained, and heat accumulates in the bearing as before. This important fact is verified by the design of high-speed bearings, which, it is always noted, are very long in proportion to their diameter, thus giving relatively high radiating power.

Bearings must be rigidly fastened to the body of the machine in some way, and the immediate support is termed a **bracket, frame,** or **housing.** "Bracket" is a very general term, and applies to the supports of other machine parts besides "bearings." It is especially applicable to the more familiar types of bearing supports, and is here introduced to make the analysis complete.

The bracket must be strong enough as a beam to take the side load, the bending moment being figured at such points as are necessary to determine its outline. It may be of solid, box, or ribbed form, the latter being the most economical of material, and usually permitting the simplest pattern. The fastening of the bracket to the main body of the machine must be broad to give stability; the bolts act partly in shear to keep the bracket from sliding along its base, and partly in tension to resist its tendency to rotate about some one of its edges, due to the side pull of the belt, gear tooth, or lever load, as the case may be. The weight of the bracket itself and of the parts it sustains through the bearing, has likewise to be considered; and this acts, in conjunction with the working load on the bearing, to modify the direction and magnitude of the resultant load on the bracket and its fastening.

Stands are forms of brackets, and are subject to the same analysis. The distinction is by no means well defined, although we usually think more readily of a stand as having an upright or inverted position with reference to the ground. The ordinary "hanger" is a good example of an inverted stand; and the regular "floor stand," found on jack shafts in some power houses, is an example of the general class.

THEORY. As the method of calculation of the diameter of the shaft, as well as its deflection, has been considered under "Shafts," we may assume that the theoretical study of bearings starts on a given basis of shaft diameter D. The main problem then being one of heat control, let us first calculate the amount of heat developed in a bearing by a given side load. The force of friction acts at the circumference of the shaft, and is equal to the coefficient of friction times the normal force; or, for a given side load P, Fig. 75, the force of friction would be μP. The peripheral speed of the shaft for N revolutions per minute is $\frac{\pi DN}{12}$ feet per minute. As work is "force times distance," the work wasted in friction is then $\frac{\mu P \pi DN}{12}$ foot-pounds per minute. One horse-power being equal to 33,000 foot-pounds per minute, we have the equation,

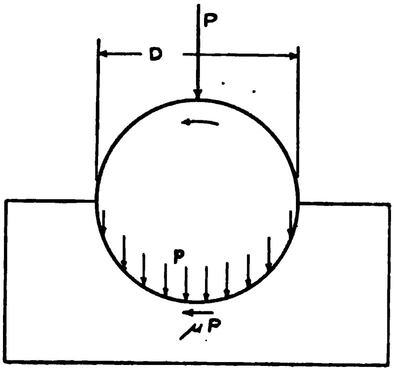


Fig. 75.

$$H = \frac{\mu P \pi DN}{12 \times 33{,}000}. \qquad (113)$$

The value of μ for ordinary, well-lubricated bearings, may run as low as 5 per cent; but as the lubrication is often impaired, it quite commonly rises to 10 or 12 per cent. A value of 8 per cent is a fair average. This amount of horse-power is dissipated through the bearing in the form of heat. If we could exactly determine the ability that each particle of the metal around the shaft had to transmit the heat, or to pass it along to the outside of the casting, and if we could then determine the ability of the particles of air surrounding the casting to receive and carry away this heat, we could calculate just such proportions of the bearing and its casing as would never choke or retard this free transfer of heat away from the running surface.

Such refined theory is not practical, owing to the complicated shapes and conditions surrounding the bearing. The best that we can do is to say that for the usual proportions of bearings the side load may exist up to a certain intensity of "pressure per square inch of projected area" of bearing, or, in form of an equation,

$$P = pLD. \qquad (114)$$

The constant p is of a variable nature, depending on lubrication, speed, air contact, and other special conditions. For ordinary bearings having continuous pressure in one direction, and only fair lubrication, 400 to 500 is an average value. When the pressure changes direction at every half-revolution, the lubricant has a better chance to work fully over the bearing surface, and a higher value is permissible, say, 500 to 800. In locations where mere oscillation takes place, not continuous rotation, and reversal of pressure occurs, as on the cross-head pin of a steam engine, p may run as high as 900 to 1,200. On the crank pins of locomotives, which have the reversal of pressure, and the benefit of high velocity through the air to facilitate cooling, the pressures may run equally high. On the eccentric crank pins of punching and shearing machines, where the pressure acts only for a brief instant and at intervals, the pressure ranges still higher without any dangerous heating action.

When a bearing, for practical reasons, is provided with a cap held in place by bolts or studs, the *theory* of the cap and bolts is of little importance, unless the load comes directly against the cap and bolts. Except in the latter case, the proportions of the cap and the size of the bolts are dependent upon general appearance and utility, it being manifestly desirable to provide a substantial design, even though some excess of strength is thereby introduced.

For the worst case of loading, however, which is when the cap is acted upon by the direct load, such as P in Fig. 76, we have the condition of a centrally loaded beam supported at the bolts. It is probable that the beam is partially fixed at the ends by the clamping of the nut; also that the load P, instead of being concentrated at the center, is to some extent distributed. It is hardly fair to assume the external moment equal to $\frac{Pa}{8}$ or $\frac{Pa}{4}$, the one being too small, perhaps, and the other too large. It will be reasonable to take the external moment at $\frac{Pa}{6}$, in which case, equating the external moment to the internal moment of resistance,

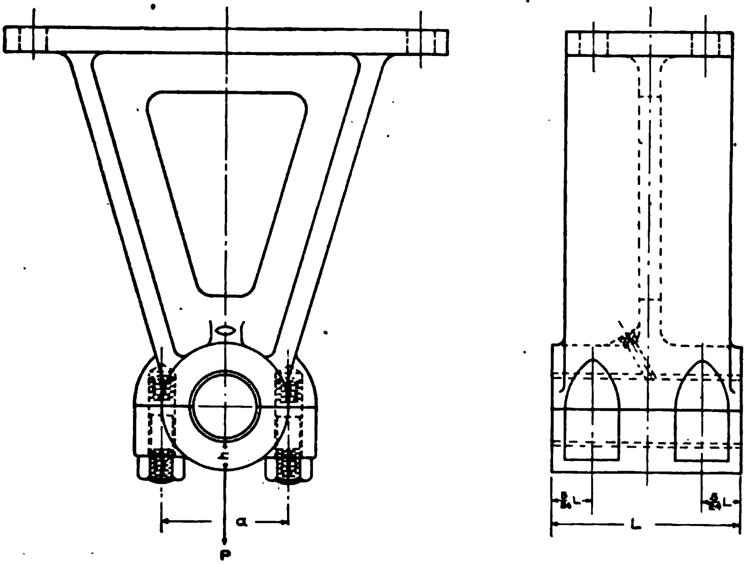


Fig. 76.

$$\frac{Pa}{6} = \frac{SI}{c} = \frac{SLh^2}{6}, \qquad (115)$$

from which, the length of bearing being known, we may calculate the thickness h.

One bolt on each side is sufficient for bearings not more than 6 inches long, but for longer bearings we usually find two bolts on a side. The theoretical location for two bolts on a side, in order that the bearing may be equally strong at the bolts and at the center of the length, may be shown by the principles of mechanics to be $\frac{5}{24}$ L from each end, as indicated in Fig. 76.

The bolts are evidently in direct tension, and if equally loaded would each take their fractional share of the whole load P. This is difficult to guarantee, and it is safer to consider that $\frac{2}{3}$ P may be taken by the bolts on one side. On this basis, for total number of bolts n, equating the external force to the internal resistance of the bolts, we have:

$$\frac{2}{3}P = \frac{S\pi d_1^2}{4} \times \frac{n}{2}, \qquad \textbf{(116)}$$

from which the proper commercial diameter may be readily found.

The bracket may have the shape shown in Fig. 77. The portion at B is under direct shearing stress; and if A be the area at this point, and S_s the safe shearing stress, then, equating the external force to the internal shearing resistance,

$$P = AS_s. \qquad \textbf{(117)}$$

The same shear comes on all parts of the bracket to the left of the load, but there is an excess of shearing strength at these points.

At the point of fastening, the bolts are in shear, due to the same load, for which the equation is

$$P = \frac{\pi d^2}{4} n_1 S_s. \qquad \textbf{(118)}$$

For the upper bolts, the case is that of direct tension, assuming that the whole bracket tends to rotate about the lower edge E. To find the load T on these bolts, we should take moments about the point E, as follows:

$$PL_1 = Tl; \text{ or } T = \frac{PL_1}{l}. \qquad \textbf{(119)}$$

Then, equating the external force to the internal resistance,

$$T = \frac{PL_1}{l} = \frac{\pi d_1^2}{4} \times \frac{n_1}{2} S. \qquad \textbf{(120)}$$

The upper flange is loaded with the bolt load T, and tends to break off at the point of connection to the main body of the bracket, the external moment, therefore, being Tr. The section of the flange is rectangular; hence the equation of external and internal moments is:

$$\mathrm{T}r = \frac{\mathrm{PL}_1}{l} r = \frac{\mathrm{S}bt^2}{6}. \qquad (121)$$

It may be noted that the lower bolts act on such a small leverage about E, that they would stretch and thus permit all the load to be thrown on the upper bolts; this is the reason why they are not subject to calculation for tension.

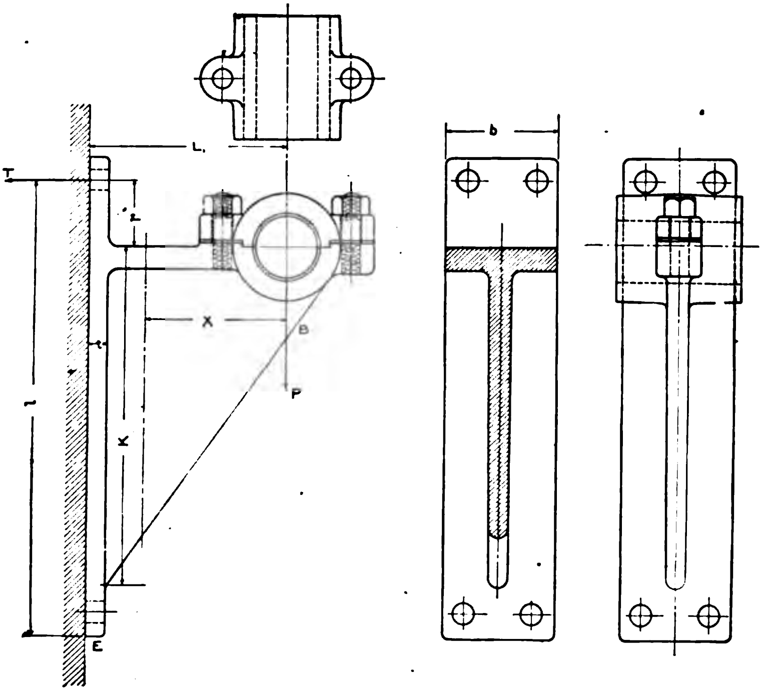


Fig. 77.

The section of the bracket to the left of the load P is dependent upon the bending moment, for, if this section is large enough to take the bending moment properly, the shear may be disregarded. It should be calculated at several points, to make sure that the fiber stress is within allowable limits. The general expression for the equation of moments is, for any section at leverage x,

$$\mathrm{P}x = \frac{\mathrm{SI}}{c}, \qquad (122)$$

from which, by the proper substitution of the moment of inertia of the section, the fiber stress can be calculated. The moment of inertia for simple ribbed sections can be found in most handbooks. The process of solution of the above equation, though simple, is apt to be tedious, and is not considered necessary to illustrate here.

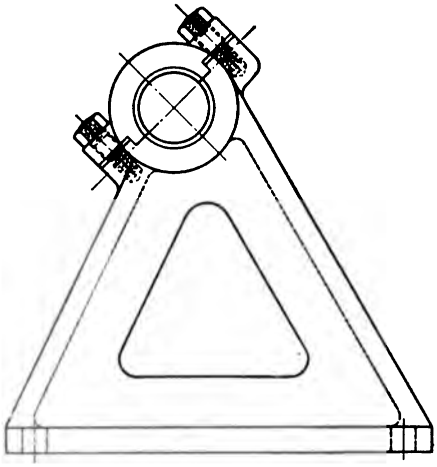


Fig. 78.

PRACTICAL MODIFICATION. Adjustment is an important practical feature of bearings. Unless the proportions are so ample that wear is inappreciable, simple and ready adjustment must be provided. The taper bushing, Fig. 79, is neat and satisfactory for machinery in which expense and refinement are permissible. This is true of some machine tools, but is not true of the general "run" of bearings. The most common form of adjustment is secured by the plain cap (which may or may not be tongued into the bracket), with liners placed in the joint when new, which may subsequently be removed or reduced so as to allow the cap to close down upon the shaft. Several forms of cap bearings are illustrated in Figs. 80, 81, and 82.

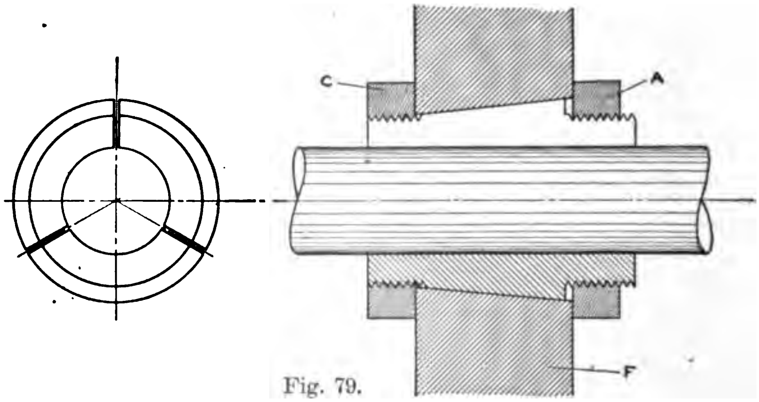


Fig. 79.

Large engine shaft bearings have special forms of adjustment by means of wedges and screws, which take up the wear in all directions, at the same time accurately preserving the alignment of the shafts; but this refinement is seldom required for shafts of ordinary machinery.

In cases where the cap bearing is not applicable, a simple bushing may be used. This may be removed when worn, and a new one inserted, the exact alignment being maintained, as the outside will be concentric with the original axis of shaft, regardless of the wear which has taken place in the bore.

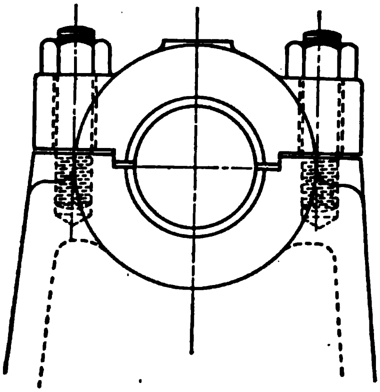


Fig. 80.

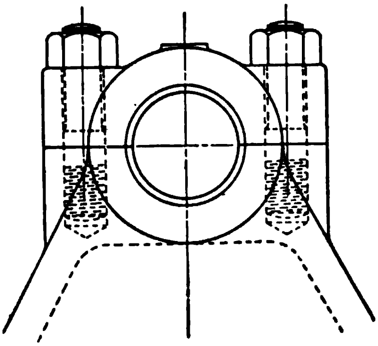


Fig. 81.

The lubrication of bearings is a part of the design, in that the lubricant should be introduced at the proper point, and pains taken to guarantee its distribution to all points of the running surface. The method of lubrication should be so certain that no excuse for its failure would be possible. Grease is a successful lubricator for heavy loads and slow speeds, oil for light loads and high speeds.

In order to insure the lubricant reaching the sliding surfaces and entering between them, it must be introduced at a point

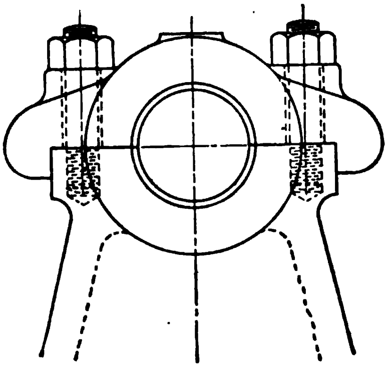


Fig. 82.

where the pressure is moderate, and where the motion of the parts will naturally lead it to all points of the bearing. Grooves and channels of ample size assist in this regard. A special form of bearing uses a ring riding on the shaft to carry the oil constantly from a small reservoir beneath the shaft up to the top, where it is distributed along the bearing and finally flows back to the reservoir and is used again.

The materials of which bearings are made vary with the service required and with the refinement of the bearing. Cast iron makes an excellent bearing for light loads and slow speeds, but it is very apt to "seize" the shaft in case the lubrication is in the least degree impaired. Bronze, in its many forms of density and hardness, is extensively used for high-grade bearings, but it also has little natural lubricating power, and requires careful attention to keep it in good condition.

Babbitt, a composition metal, of varying degrees of hardness, is the most universal and satisfactory material for ordinary bearings. It affords a cheap method of production, being poured in molten form around a mandrel, and firmly retained in its casing or shell through dovetailed pockets into which the metal flows and hardens. It requires no boring or extensive fitting. Some scraping to uniform bearing is necessary in most cases, but this is easily and cheaply done. Babbitt is a durable material, and has some natural lubricating power, so that it has less tendency to heat with scanty lubrication than any of the materials previously mentioned. Almost any grade of bearing may be produced with babbitt. In its finest form the babbitt is hammered, or pened, into the shell of the bearing, and then bored out nearly to size, a slightly tapered mandrel being subsequently drawn through, compressing the babbitt and giving a polished surface.

A combination bearing of babbitt and bronze is sometimes used. In this the bronze lies in strips from end to end of the bearing, and the babbitt fills in between the strips. The shell, being of bronze, gives the required stiffness, and the babbitt the favorable running quality.

PROBLEMS ON BEARINGS, BRACKETS, AND STANDS.

1. The allowable pressure on a bearing is 300 pounds per

square inch of projected area. What is the required length of the bearing if the total load is 4,500 pounds and the diameter is 3 inches?

2. The cross-head pin of a steam engine must be 2.5 inches in diameter to withstand the shearing strain. If the maximum pressure is 10,000 pounds, what length should be given to the pin?

3. The journals on the tender of a locomotive are $3\frac{1}{2} \times 7$ inches. The total weight of the tender and load is 60,000 pounds. If there are 8 journals, what is the pressure per square inch of projected area?

4. What horse-power is lost in friction at the circumference of a 3-inch bearing carrying a load of 6,000 pounds, if the number of revolutions per minute is 150 and the coefficient of friction is assumed to be 5 per cent?

5. The cast-iron bracket in Fig. 77 has a load P of 1,000 pounds. Determine the fiber stress in the web section at the base of the bracket if the thickness is taken at $\frac{1}{2}$ inch, and $L_1 = 12$ inches; $l = 20$ inches; $k = 11$ inches; $t = 1$ inch.

6. Calculate the diameter of the bolts at the top of the bracket.

7. Assuming r equal to 6 inches, what is the fiber stress at the root of flange?

EXAMINATION PAPER

MACHINE DESIGN.

At the end of each chapter on the several subjects in Part II will be found a series of test problems intended to cover the ground of the chapter which they terminate. As a matter of *benefit to the student*, he should try to solve each full set of problems at the time he is pursuing his detailed study of the chapter. As a matter of *formal examination*, the following selected problems are required, the student being given some latitude of choice. He must confine himself strictly within the limits stated, however, and if he submits more than the required number of problems in each set, his grade will be invariably determined on the first problems read by the instructor up to that required number.

Belts	Solve	any	3	problems
Pulleys	"	"	3	"
Shafts	"	"	5	"
Spur, Bevel, and Worm Gears	"	"	4	"
Friction Clutches	"	"	2	"
Couplings	"	"	2	"
Bolts, Studs, Nuts, and Screws	"	"	3	"
Keys, Pins, and Cotters	"	"	4	"
Bearings, Brackets, and Stands	"	"	4	"

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ook may be kept